COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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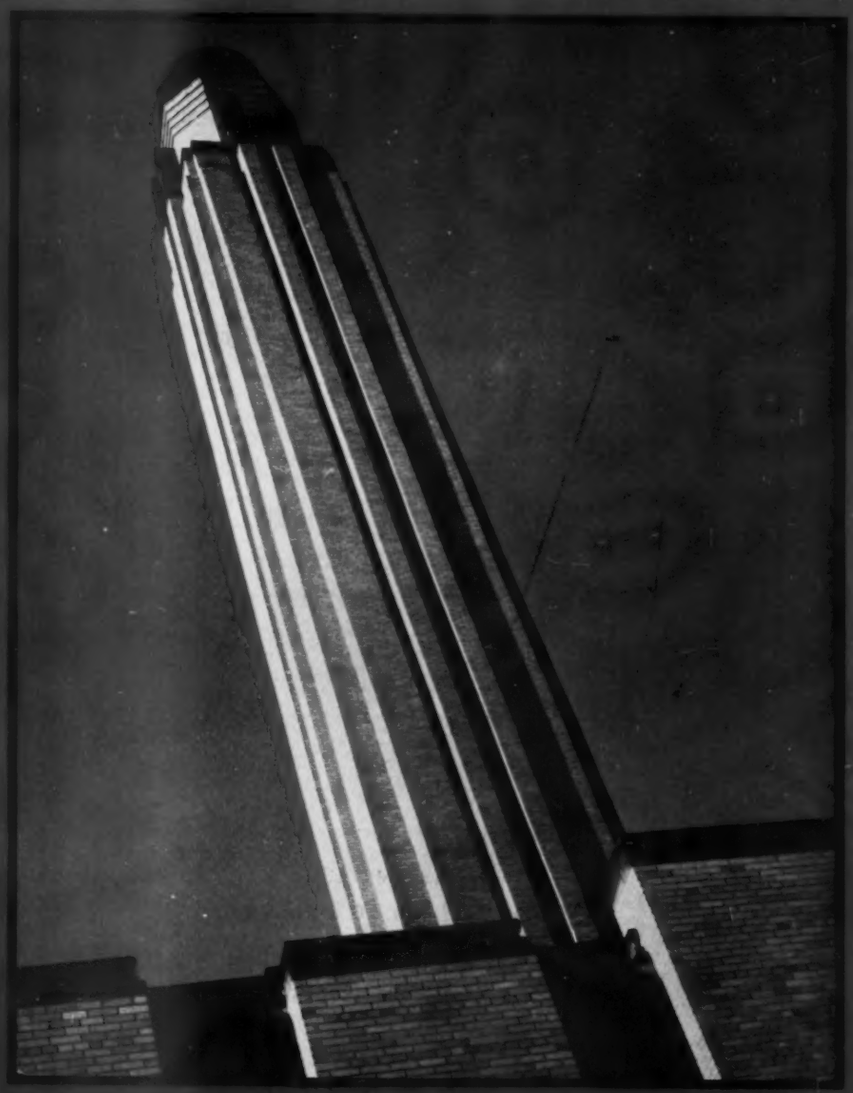

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Complete Boiler Control Maintains Balanced Operation at Des Moines

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Slag Deposits on Heat-Absorbing Surfaces and Means for Removal

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COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME ELEVEN NUMBER EIGHT

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NO. 1 IN A SERIES OF STUDIES OF BOILER WATER LEVEL CONTROL WITH COPES FLOWMATIC

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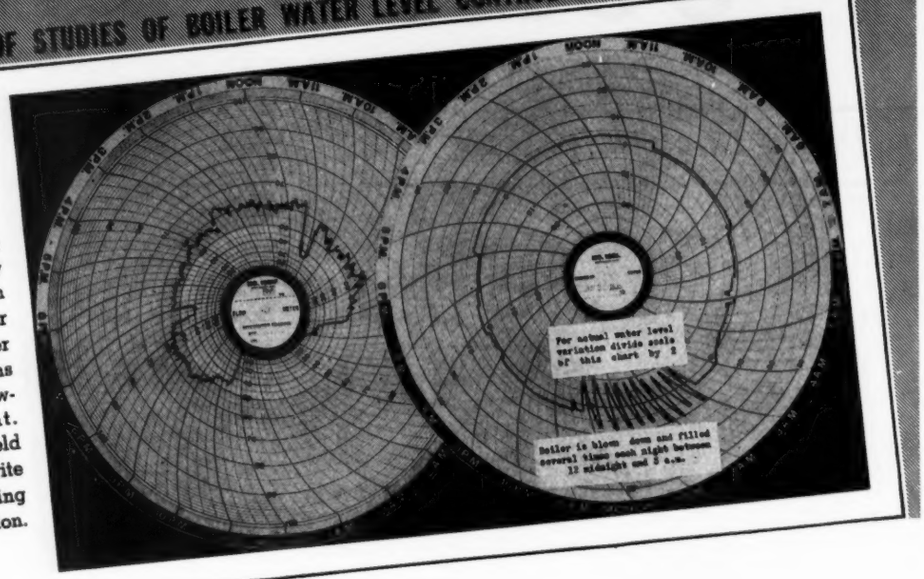
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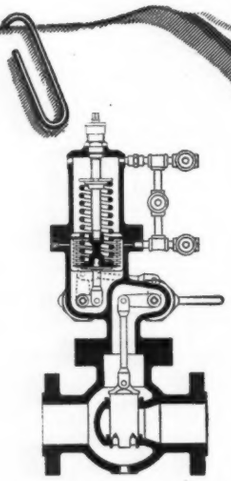

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Results in this plant emphasize two good reasons why the swing is to the COPES Flowmatic for effective feed water regulation in modern steam plants. Close water level control under all load conditions increases safety and permits higher efficiency. Trouble-free operation—this user says, "No unit in our plant has operated with less attention"—saves time in the boiler room and saves money in maintenance. The simplified steam-flow type COPES Flowmatic is giving results like these in plants from 80 to 1325 pounds W.S.P.—on boilers generating from 15,000 to 550,000 pounds per hour. Write for Bulletin 409-B.

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EDITORIAL

Seek Tax Reimbursement

A long-standing point of contention between proponents and opponents of public ownership of utilities has centered around taxation. Those comprising the latter group and most unbiased engineers, have insisted that equivalent taxes should be included in figuring the cost of publicly generated energy when making comparison with that privately produced. This is on the theory that where the publicly owned plant pays no taxes this income is lost to the community, to the state and to the nation, and must be made up in other ways. With mounting taxation, as at present, this becomes increasingly important.

It is therefore significant that certain sections of the South, which a few years ago were clamoring for TVA power at "yardstick" rates, have become alarmed at the loss in revenue formerly enjoyed through taxing the local utilities. Congress has now been approached by a delegation, headed by the Governor of Tennessee, with the plea that it make tax reimbursements to these communities for the losses thus sustained.

Apparently it was not enough for the national treasury to be heavily tapped for the cost of this regional development, but now it is proposed to ask the taxpayers at large for further gratuities. Perhaps the present so-called economy block in Congress may stand in the way.

Deslagging Operations

The deslagging of furnace walls and convection heat-absorbing surfaces has become a routine operating problem with many modern high-capacity steam-generating units, the frequency and extent depending upon the fuel characteristics, the load and the design. Some units and the conditions under which they operate require very little attention on this score, whereas with others, due to a combination of unfavorable factors, considerable labor is required to prevent excessive fouling of such surfaces.

In the report of the Prime Movers Committee on "Combustion," recently issued by the Edison Electric Institute, statements by ten operating companies indicate a wide variation in the means adopted to cope with this problem. In most of the plants covered hand lancing is employed to supplement the soot blowers, but opinions based on experience appear to differ as to the desirability of using air, steam or water as the hand-lancing medium. In several cases air has been found the most satisfactory, whereas in one it proved ineffective. Three of the stations employ water for deslagging furnace walls although steam is preferred for cleaning superheater and economizer surfaces. In other cases soot blowers alone, properly distributed and employing steam at adequately high pressure, have been sufficient.

The variation in experiences with the several deslagging methods may be partly due to differences in the character of the slag. As pointed out by Mr. Keppler in the current issue, this is governed largely by the conditions under which the suspended ash lands on the heat-absorbing surfaces, which, in turn, is affected by the ash temperature, the velocity, and both the temperature and the smoothness of these surfaces.

Many engineers are reluctant to use water for lancing because of the possible damage that may result if it is not properly applied. They feel that, despite prescribed precautions, it is not always possible to control the human element. In this connection the procedure outlined by Mr. Keppler should prove instructive.

Whatever the method employed for the removal of slag deposits, it represents an additional operating cost for labor and steam, air or water that must be taken into consideration in the selection of the fuel and be warranted by increased availability.

Engineers in Civic Affairs

That engineers should take a more active part in local civic affairs has long been advocated and discussed in society meetings, in papers and speeches, and in the technical press. Every angle of the question, *pro* and *con*, has been covered many times. As a result, considerable progress has been made in advancing the professional status of the engineer and in the placing of technically trained men in public positions of an engineering character. Also, much constructive work has been carried out by committees representing engineering societies acting in an advisory capacity to local bodies. These accomplishments have been largely through collective effort. Individually, however, engineers still appear very much in the minority as leaders in civic affairs, despite occasional outstanding examples.

This was borne out recently by the results of a nation-wide survey initiated by the American Association of Engineers, by means of questionnaires sent to Chambers of Commerce, Rotary, Kiwanis and Lions Clubs throughout the United States, all of which are important factors in community affairs. Numerous questions were asked and the replies were varied. As to the question, "How do engineers compare with other vocations in civic activities," twenty-five per cent of the replies indicated that engineers as a class had been outstanding; five per cent accorded qualified recognition; nineteen per cent stated flatly that engineers had not been useful beyond their strict line of duty; and fifty-one per cent ignored this question entirely, thus indicating that if the engineers had been sufficiently active, those replying would have been able to make a definite statement.

This is not very encouraging but would appear to indicate that engineers prefer to exert their influence collectively rather than individually.

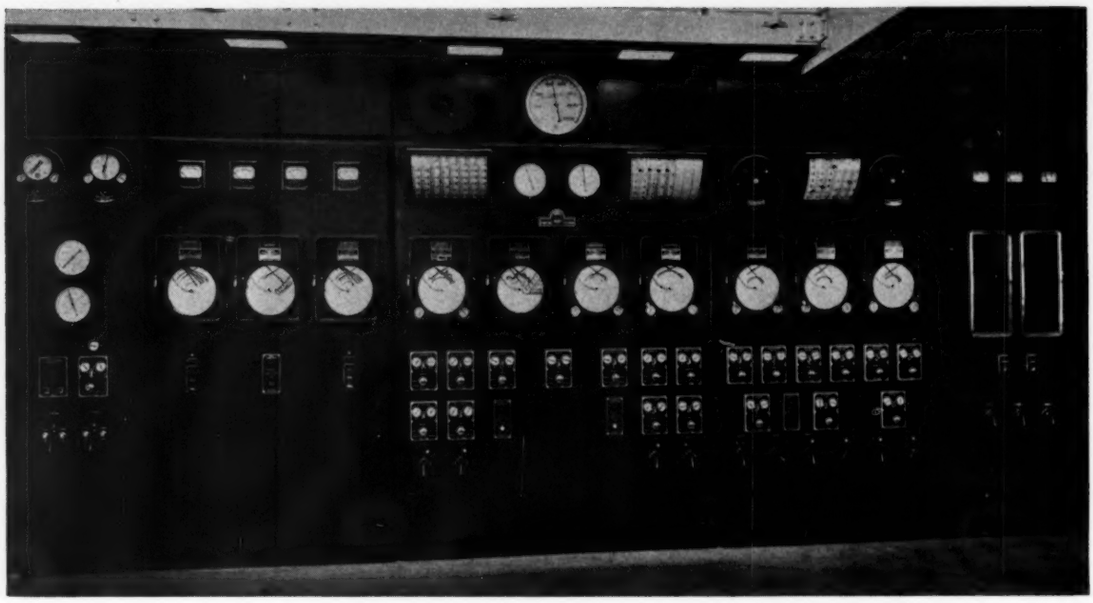


Fig. 1—Automatic control of combustion, feedwater flow, steam temperature, pressure reducing, desuperheating and coal-air temperature is centered on this boiler-control panel

COMPLETE BOILER CONTROL
Maintains Balanced Operation at Des Moines

By J. F. McLAUGHLIN, Mech. Engr.
Iowa Power and Light Company

THE more complete an automatic control system is made, the more opportunity there is to derive the maximum efficiency from the operation which it controls. This is particularly true in the case of large, high-pressure and high-temperature boilers which have become common in central stations during recent years. In these, any deficiency, due to lack of control over factors that are not particularly important in smaller installations, is likely to cause considerable loss. This loss may not occur in the uncontrolled portion of the system, but in other portions due to upset conditions caused by unbalance in the uncontrolled section.

All this was kept in mind by those responsible for planning the recently added Plant No. 2 of the Iowa Power and Light Company. Located at Des Moines, Iowa, the new addition includes a 420,000-lb per hr capacity pulverized-coal-fired boiler. This unit of the bent-tube type, converts 400 F feedwater into steam at 1340 lb per sq in. and 925 F total steam temperature, the steam being utilized in a new 35,000-kw turbine.

Automatic combustion control was unquestionably desirable, and with such operating conditions it was seen that complete control of all important operating factors would contribute largely to the efficiency of the entire system. Thus, automatic control of feedwater flow, steam temperature, pressure reducing, desuperheating and coal-air temperature, in addition to combustion control, was included in the specifications. The control system selected was of the Bailey air-operated type.

In this high-pressure, high-temperature boiler installation supplying steam to a 35,000-kw turbine, combustion control is supplemented by automatic control of the feedwater, pressure reducing and desuperheating, the steam temperature and temperature of the mixture of cold and preheated air supplied to the pulverizers.

Combustion Control

The combustion control layout is shown diagrammatically in Fig. 2 and consists of a master steam-pressure recorder which controls the rate of raw coal feed and the position of the exhauster inlet damper. Steam pressure also serves as the primary controlling factor for the induced-draft and forced-draft fans. The steam flow-air flow relationship serves as a secondary or repositioning control of the induced-draft fan and the forced-draft fan, while further control over the forced-draft fan, by compensating its position in accordance with the furnace draft, is made possible through the furnace-draft controller. Thus, truly complete automatic combustion control is provided through this closely coordinated regulation of fuel and air with the initial control dependent upon the steam pressure and with the steam flow-air flow relationship giving automatic readjustment.

Feedwater Control

The feedwater control is of the three-element type which is particularly well suited to boilers with high steaming rates and also those with once-through flow

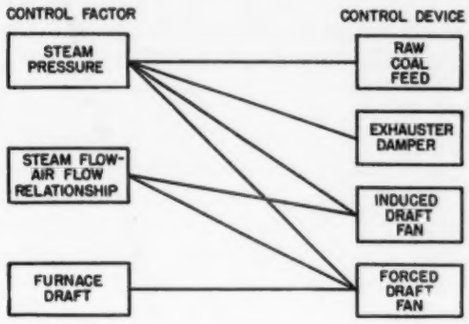


Fig. 2—Diagram of combustion control

characteristics. Shown diagrammatically in Fig. 3 it operates as follows:

The feedwater flow to the boiler and the steam output are measured by meters on the panel and these readings are balanced against each other by means of a differential linkage. The air pilot valve, operated by this differential linkage, sends out a corrective loading pressure whenever the rate of feedwater input drops below or rises above the rate of steam output. This loading pressure, applied through a relay, repositions the feedwater control valve to restore a balance between water input and steam output. An air pilot valve operated by the boiler-drum water-level recorder makes any readjustment that may be necessary in the rate of feedwater flow to maintain the proper water level. This is done by creating a loading pressure which modifies that of the steam flow-water flow differential in the averaging relay before it is applied to the valve.

Pressure-Reducing and Desuperheating Control

Designed for future topping operation this control feature is as follows:

All during the time that the new high-pressure turbine is in operation, the bypass in which the pressure-reducing and desuperheating valve is located, is shut off. (See diagrammatic layout Fig. 4.) The turbine exhausts into the low-pressure section of the plant and there is no need for such reducing apparatus.

However, if for any reason the turbine should go off the line, a quick-opening valve immediately bypasses all the steam through the reducing valve. This reducing

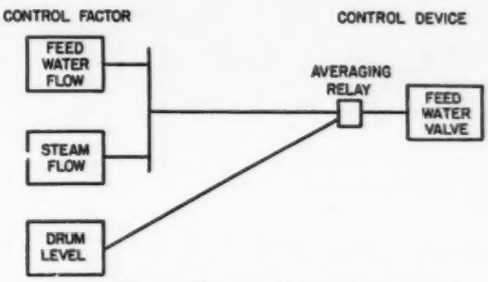


Fig. 3—Three-element feedwater control

valve is continuously positioned, during turbine operation, by the turbine steam-flow recorder-controller so that it will be open the proper amount at any time the quick-opening valve directs the steam through it.

When the steam is started through the reducing valve, the turbine steam-flow recorder-controller, which initially positioned it, loses all control effect inasmuch as no steam

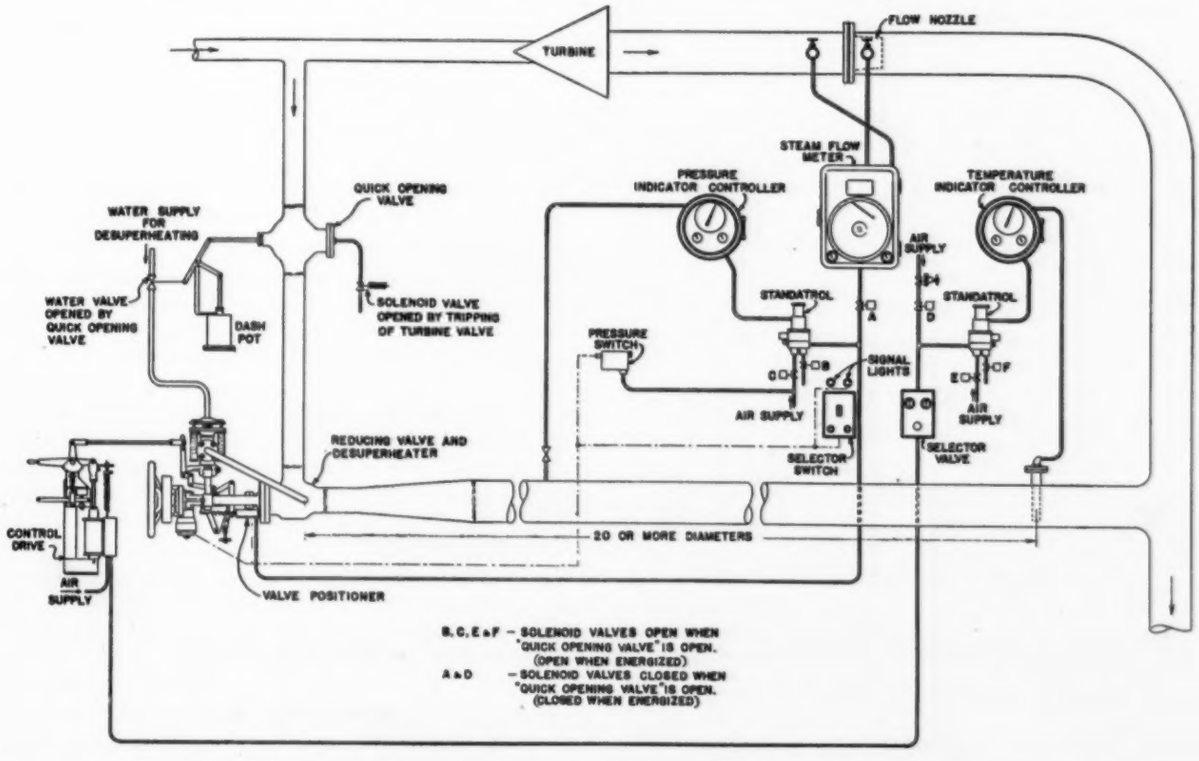


Fig. 4—Pressure-reducing and desuperheat control layout

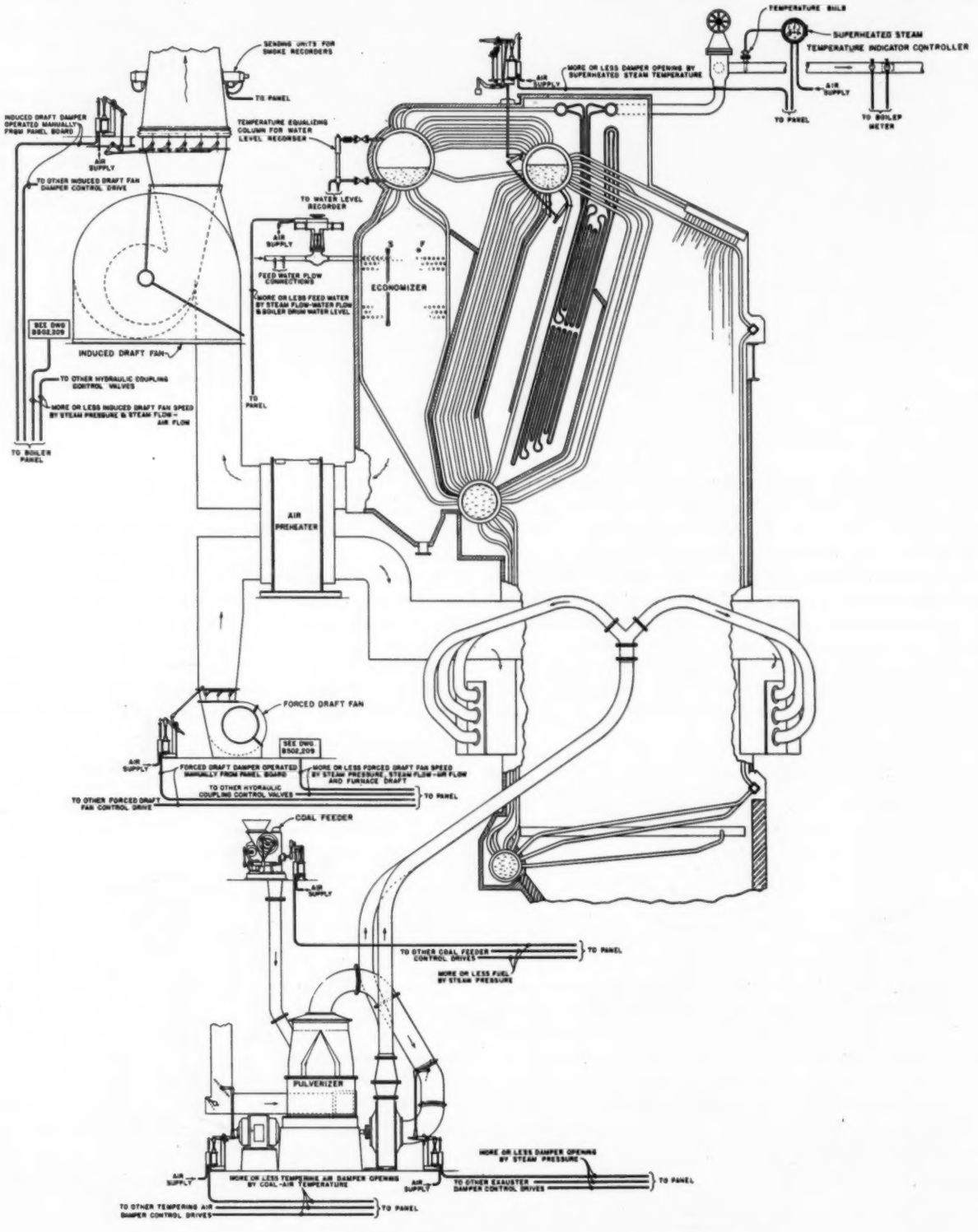

TEMPERATURE BULB
SUPERHEATED STEAM
TEMPERATURE INDICATOR CONTROLLER
SENDING UNITS FOR SMOKE RECORDERS
TO PANEL
MORE OR LESS DAMPER OPENING BY SUPERHEATED STEAM TEMPERATURE
AIR SUPPLY
TO PANEL
TO BOILER METER
INDUCED DRAFT DAMPER OPERATED MANUALLY FROM PANEL BOARD
AIR SUPPLY
TEMPERATURE EQUALIZING COLUMN FOR WATER LEVEL RECORDER
TO OTHER INDUCED DRAFT FAN DAMPER CONTROL DRIVE
TO WATER LEVEL RECORDER
AIR SUPPLY
FEED WATER FLOW CONNECTIONS
ECONOMIZER
MORE OR LESS FEED WATER BY STEAM FLOW-WATER FLOW & BOILER DRUM WATER LEVEL
SEE DWG 9502,209
INDUCED DRAFT FAN
TO OTHER HYDRAULIC COUPLING CONTROL VALVES
MORE OR LESS INDUCED DRAFT FAN SPEED BY STEAM PRESSURE & STEAM FLOW - AIR FLOW
TO BOILER PANEL
TO PANEL
AIR PREHEATER
FORCED DRAFT FAN
SEE DWG 9502,209
AIR SUPPLY
FORCED DRAFT DAMPER OPERATED MANUALLY FROM PANEL BOARD
MORE OR LESS FORCED DRAFT FAN SPEED BY STEAM PRESSURE, STEAM FLOW-AIR FLOW AND FURNACE DRAFT
TO OTHER HYDRAULIC COUPLING CONTROL VALVES
TO PANEL
TO OTHER FORCED DRAFT FAN CONTROL DRIVE
COAL FEEDER
AIR SUPPLY
TO OTHER COAL FEEDER CONTROL DRIVES
TO PANEL
MORE OR LESS FUEL BY STEAM PRESSURE
PULVERIZER
MORE OR LESS DAMPER OPENING BY STEAM PRESSURE
AIR SUPPLY
MORE OR LESS TEMPERING AIR DAMPER OPENING BY COAL-AIR TEMPERATURE
AIR SUPPLY
TO OTHER EXAUSTER DAMPER CONTROL DRIVES
TO PANEL
TO OTHER TEMPERING AIR DAMPER CONTROL DRIVES
TO PANEL

Fig. 5—Layout of C-E steam-generating unit with bowl mills showing locations of various Bailey control drives and the points of control

will be passing through the turbine line. The control of the pressure reduction is, therefore, immediately assumed by the pressure indicator-controller which maintains the proper outlet pressure.

The desuperheat control is carried out in the same manner. The instant the quick-opening valve bypasses the steam, water is injected into the reducing valve to cool the steam. This initial flow of water is also determined ahead of time by constant positioning through the turbine-flow recorder-controller. As soon as the latter drops out of the control setup the temperature indicator-controller, with its sensitive element in the low-pressure line, takes over and controls the desuperheater water flow until the turbine is again put on the line.

Until such time as the topping turbine is installed, the quick-opening valve can be opened by the fireman, and steam for the low-pressure turbines can be furnished from the high-pressure boiler through the desuperheater, the pressure and temperature being controlled as described.

Steam-Temperature Control

Control of the temperature of the superheated steam is provided for by means of a bypass damper at the entrance to the superheater section. If the temperature of the superheated steam exceeds that desired, the air pilot valve in the steam-temperature controller sets up a loading pressure, which, through a "Standatrol" relay, causes the control drive to open the bypass damper and permit the flue gases to pass, in part, through an open pass beyond the superheater section baffle. The amount that the damper is opened determines how much or how little of the flue gases will pass over the superheater section due to the lower resistance of the open pass.

Coal-Air Temperature Control

Both from the standpoint of safety and economy, it was deemed advisable to apply automatic control to the mixing of preheated and cold air supplied to the pulverizers, in order to maintain a uniform temperature of the coal and air mixture. This is accomplished by means of a coal-air temperature recorder-controller for each pulverizer. The controller sets up an air-loading pressure which causes a control drive to position the tempering-air damper in the duct carrying hot air from the preheater to the pulverizer. A hand-set damper in the cool-air duct is locked in place in a position giving a good operating characteristic to the damper in the hot-air duct. Thus, control of the hot air admitted to the pulverizer also controls, simply but accurately, the temperature of the coal-air mixture.

In addition to the main controlling systems which have been briefly outlined, there are other automatic features, such as boiler-purge interlocks and control to prevent recirculation of the air when only one of two forced-draft fans is being used, which contribute considerably to the completeness of the installation. With all of the principal portions of the system controlled, there are no upset portions to affect the operation of any other part. This permits the various control systems to operate hand-in-hand as a unit, continuously maintaining the desired conditions.

To date the operation of the control has been very satisfactory, leading us to believe that all of the control supplied is both desirable and helpful.

A.S.M.E Spring Meeting at Worcester, Mass.

The 1940 Spring Meeting of the American Society of Mechanical Engineers will be held in Worcester, Mass., on May 1 to 3, with headquarters at the Hotel Bancroft. During the meeting the Worcester Section of the Society will celebrate the twenty-fifth anniversary of its founding. A diversified program has been prepared including sessions on iron and steel, machine-shop practice, process industries, management, materials handling, heat transfer, power, fuels, boiler feedwater and hydraulics. Papers to be presented at the five last mentioned sessions are as follows:

WEDNESDAY, May 1

10 a.m.

POWER SESSION

Feedwater Treatment, by C. H. Fellows of the Detroit Edison Co.

Industrial Steam Power, by W. F. Ryan of Stone & Webster Engineering Corp.

8 p.m.

FUELS SESSION

Burning of Heavy Fuel Oil, by G. C. Martinson, Sinclair Refining Co.

Motion Pictures on Furnace Operation, by

1. A representative of the Consolidated Edison Co. of New York
2. O. F. Campbell, Sinclair Refining Co.
3. Otto de Lorenzi, Combustion Engineering Co.
4. Prof. S. Konzo, University of Illinois

THURSDAY, May 2

9:30 a.m.

FUELS SESSION

Purchase and Use of Fuels, by E. W. Stone, Bigelow-Sanford Carpet Co.

Effect of Coal Characteristics on Pulverized Coal Firing, by Ollison Craig, Riley Stoker Corp.

Recent Developments of Oil Burners, by R. C. Vroom, Peabody Engineering Corp.

BOILER FEEDWATER SESSION

Determination of the Purity of Steam, by M. C. Schwartz, W. B. Gurney, and T. E. Crosson, Gulf States Utilities Co.

Discussion of paper by R. E. Hall and E. P. Partridge on the Attack on Steel in High-Pressure Boilers as a Result of Overheating Due to Steam Blanketing. (This paper was originally presented at the 1939 Spring Meeting in New Orleans)

HYDRAULIC SESSION, I (AT ALDEN LABORATORY)

Low Velocity Characteristics of the Salt-Velocity Method, by L. J. Hooper, Worcester Polytechnic Institute

Comparative Tests of Flow Measurements, by O. H. Dodkin, Brazil, S. A.

Inspection of Alden Hydraulic Laboratory

FRIDAY, May 3

9:30 a.m.

HEAT TRANSFER SESSION

Thermodynamic Properties of Vapors, by Erich Leib, Combustion Engineering Co.

New Specific Heats, by Prof. R. C. H. Heck, Rutgers Univ.

Properties of Hydrogen Mixtures, by A. W. Brunot, General Electric Co.

HYDRAULIC SESSION, II

Progress Report on Cavitation Testing of Materials, by B. G. Rightmire, Massachusetts Institute of Technology

Cavitation Tests of Hydraulic Turbines, by R. E. B. Sharp, Baldwin-Southwark Div., Baldwin Locomotive Wks.

The program also includes a number of inspection trips to local industries as well as social features.

TURBINE-GENERATORS of Small and Medium Size

By RONALD B. SMITH
Development Engineer, Elliott Co.

Turbine-generators up to 7500 kw for steam conditions not exceeding 650 lb and 825 F are discussed. Developments leading to the present improved economic performance and operating reliability are reviewed and comparisons are made with the efficiencies of larger machines. Corresponding costs are dealt with and the distribution of losses analyzed, particularly with reference to means for effecting moisture separation within the machine. A typical unit is described.

IN THE steam turbine field the most spectacular developments have been on large machines where the financial incentive toward more economical operation has been greatest from the customer's standpoint, and the financial return to the manufacturer has been sufficiently large to cover his design and research activities. However, the developments in turbine-generators in ratings up to 7500 kw have not lagged. Designs for operating conditions up to 600 lb per sq in. and 800 F are now common, and at least one manufacturer is prepared to offer designs for service at 900 F and with pressures in excess of 1200 lb. Economic studies show that, for the present at least, it is only in rare instances that throttle pressures in excess of 650 lb can be justified in plants having units of 5000 kw capacity, inasmuch as the cost of auxiliary equipment and of auxiliary power, particularly for the boiler-feed pumps, becomes overly important items. Since it is also questionable whether the operating personnel in the average small plant is anxious at present to handle extreme temperatures, it seems fairly safe to predict that for the next few years the major activity on the part of builders of turbine equipment up to 7500 kw will be confined within 650 lb and 825 F.

The last twenty years have seen many changes in the internal design of small turbine-generators—all pointing toward increased efficiency. In the main, these have been toward additional staging, more effective nozzles and buckets, and increased attention to such details as gland leakage, rotation loss and moisture removal. In Fig. 1 is shown the trend in Rankine cycle efficiency since 1916. These data are referred to steam conditions of 200 lb, 200 deg F superheat and a vacuum of 28 in. In Fig. 2 are shown the efficiencies at their best point for central-station turbines with a rating of 30,000 kw., operating at the same steam conditions.[1] Since 1916 there has been an improvement of about 17 per cent in the efficiency of a 100-kw set, about 12½ per cent on a 5000-kw set, and about 6 per cent on a 30 000-kw machine. Over the same period generator efficiencies have changed only slightly. The fact that small machines have been able to show the larger gain is accounted for by added staging. For large central-station units, on

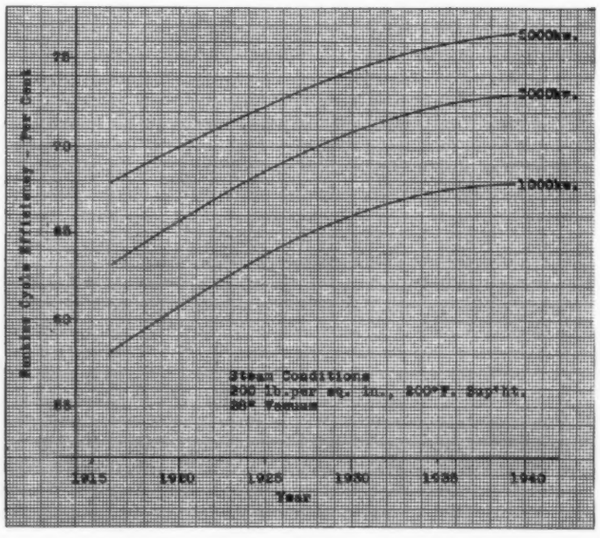


Fig. 1—Approximate full-load efficiency of small turbine generator units

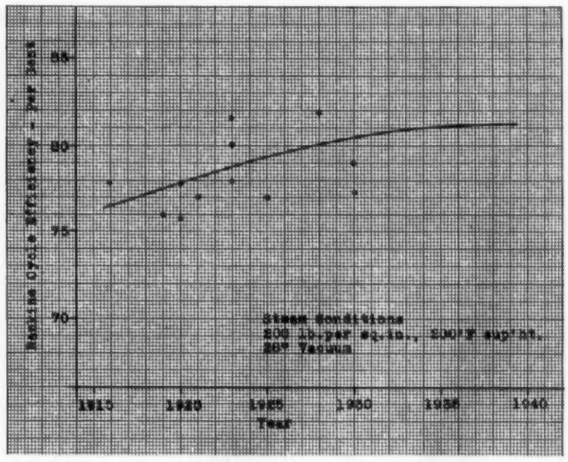


Fig. 2—Approximate maximum efficiency of 30,000-kw turbine-generators

[1] These data are secured from reported acceptance tests on units of 30,000 kw rating and are corrected to a throttle condition of 200 lb pressure, 200 deg F superheat and a vacuum of 28 in. Except for a possible further improvement from the use of hydrogen-cooled alternators they are probably fairly representative of modern practice.

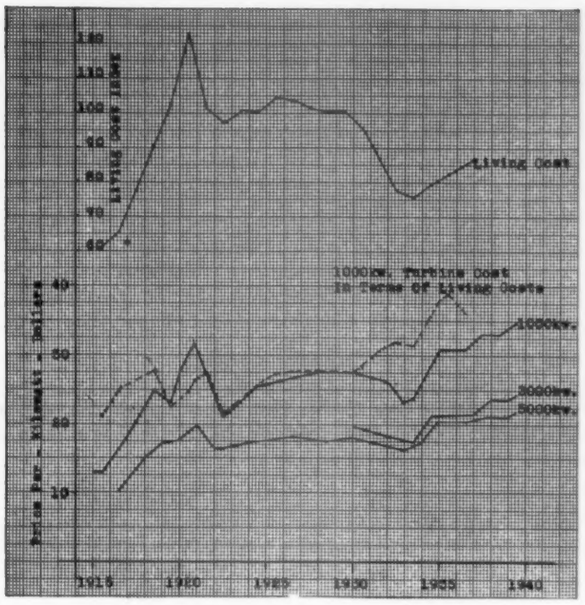


Fig. 3—Cost trend of turbine-generators

the other hand, economy and price considerations have always been important enough to warrant the most effective staging. In part then, this increase in efficiency on small machines has been effected by added equipment and cost and this, in turn, is reflected to some extent in the purchase price.

The improvement in the efficiency of small machines has resulted in a reduced differential in the cost—at least as far as fuel is concerned—between a kilowatt-hour of energy generated in a modern small plant and in a modern central station. For throttle steam conditions of 550 lb and 800 F with a vacuum of 28 in., a modern turbine having a capacity of 30,000 kw or more will generate a kilowatt-hour with about 11,500 Btu; whereas a 5000-kw plant will require for similar conditions about 12,400 Btu, or 8 per cent more. These rates refer to the turbine alone and are altered to some extent when extraction feed heating is considered.

Large stations are usually designed for four stages of feedwater heating, whereas small stations can economically justify only two; so that with this consideration, which results in a correction favoring the central station, the difference in cost of fuel per kilowatt-hour is about $10^1/_2$ per cent. When the efficiency of the boilers and auxiliary equipment is considered, as well as the more favorable load distribution arrangements that can be made on a large system, it is probably fair to state that a small plant may generate energy at a fuel cost not necessarily more than 15 per cent greater than a modern utility. Stated in a more recognizable way, whereas a utility generates a kilowatt-hour of electrical energy with one pound of coal, a 5000-kw turbine will require 1.15 lb of coal. This difference is not great—in fact it is nearly insignificant when compared to the more important items such as capital charges and distribution cost that go to make up the ultimate price of energy.

While the efficiency of small turbine-generators has progressed in a distinctly favorable way in the last 20 years, the situation in respect to costs is more involved. In Fig. 3 is shown the price per kilowatt for turbine-generator sets of 1000- and 5000-kw ratings. As a result of fluctuating market conditions and limited data these curves represent merely a trend. Consider the case of a 1000-kw machine, which since 1915 has risen in price from \$13 per kw to about \$35 per kw. Restricting the discussion to the period from 1915 to 1936, within which statistical data on the price of other commodities are available, it appears from an analysis made by the National Industrial Conference Board[2] that the cost of living over this interim has increased from 61 to 85 or about 40 per cent. The cost of a 1000-kw. turbine in terms of the cost of living,[3] as shown on Fig. 3 indicates a change from 1915 to 1936 from \$21.30 per kw to \$36.30 per kw. Through this same period, turbine efficiencies have risen as shown in Fig. 1. On 1000-kw units this has been about $17^1/_2$ per cent, which on a plant having a capacity factor of about 45 per cent and with coal at \$3.50 per ton would have amounted to a yearly saving of \$2120. Capitalized at the rate of 12 per cent, the improvement in efficiency between 1915 and 1936 has been worth a premium of about \$17.60 a kilowatt. Considering this feature in connection with the living cost index, it is apparent that a modern 1000-kw turbine-generator might legitimately cost \$38.90 on the basis of the 1915 values; instead it cost \$36.30, which represents a saving of about 8 per cent on a more or less "tailor made" product. On the surface, this saving is discouragingly small; actually hidden beneath it are improvements from the use of more costly and better materials, additional auxiliary equipment, multi-valve governing systems, etc., all of which represent increased quality to the purchaser, yet additional expense to the manufacturer.

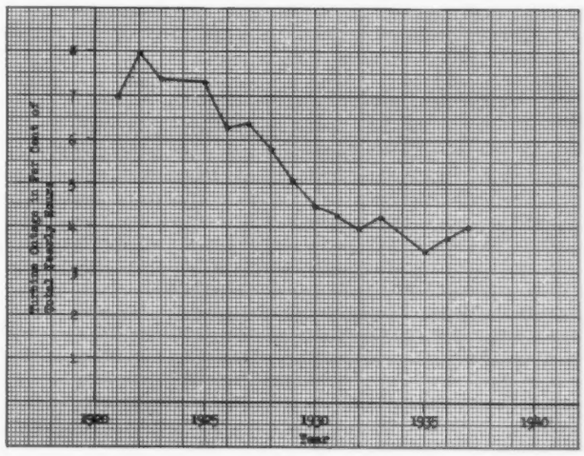


Fig. 4—Outage record of large turbine-generators

Two important considerations, namely, economical performance and operating reliability, have always directed development in turbine-generators, both large and small. Operating records on small turbine-generator sets are seldom available, but Fig. 4 shows the outage record in per cent of the year's operation for turbine-generators of 20,000-kw capacity and greater as reported by the Edison Electric Institute.[4] Inasmuch as these data always include the records of the most advanced central-station machines, it is reasonable to sug-

[2] "Cost of Living in the United States" by M. A. Beney, National Industrial Conference Board, Inc., New York, N. Y., 1937.
[3] This value is obtained by dividing actual turbine costs in cents per kilowatt by the index of living costs. This index is based on a value of 100 in 1923.
[4] "Turbines, Condensers and Pumps, 1937," Edison Electric Institution, New York, N. Y.

gest that the service record for smaller sets is even more favorable, inasmuch as less hazardous conditions are encountered. Since the early 1920's, the outage, as a result of turbine difficulty, has been decreased from about 8 per cent to about 5 per cent of the operating time. This represents a percentage decline of more than 35 per cent. The most common reason for outage is annual inspection and overhaul which in all years has constituted between 50 and 60 per cent of the whole. The second major item is blade failure, a condition only infrequently encountered on small 3600-rpm sets. Other outage sources of importance are the control gear, lubricating system and the disks or spindles. All these features are being subjected to continued engineering analysis for the purpose of eliminating operating hazards and ineffective design details.

Evaluation of Increased Economy

While operating reliability is of utmost importance and the lack of it inexcusable in turbines conservatively designed for moderate steam conditions, economy of operation is still the most frequently discussed characteristic concerned with the purchase and the operation of the machine. A 5000-kw turbine operating at a load factor of 45 per cent, and with modern steam conditions, will burn in its normal life coal to the value of from six to ten times the original cost of the turbine-generator unit. To put the problem more realistically, each per cent improvement in efficiency on a small machine secured either by design or by effective operation is worth possibly 3 per cent additional outlay on the turbine-generator investment. In view, then, of its importance, let us inquire into the manner in which the losses are distributed in a modern condensing turbine of the type illustrated by Fig. 5.

This is a 4000-kw machine for steam conditions of 400 lb per sq in., 700 F and 28½-in. vacuum. The machine is of the impulse design and consists of a single Curtis-type wheel on the governing end followed by twelve Rateau-type stages. The overall efficiency of this unit is about 76 per cent at full load, and with a combined generator and mechanical efficiency of about 96 per cent, this means an internal efficiency of 79.5 per cent for the turbine alone. Actually as a result of the reheat gain the internal losses are larger than the 20.5 per cent difference between 100 per cent and the internal efficiency of the machine. For this turbine the reheat was such as to necessitate an accounting of about 26 per cent loss, which arises mainly as a result of friction in the nozzles and buckets, leakage, moisture, windage losses on the rotating disks, a leaving loss, and a throttling loss through the inlet and governor valves. These components are distributed as follows:

	Per cent
Bucket losses	10.5
Nozzle losses	7.0
Moisture losses	2.8
Windage losses	1.8
Leaving loss	1.8
Interstage-gland losses	0.6
High- and low-pressure gland losses	0.8
Throttle loss	0.7
Total,	26.0

This distribution, which is fairly representative of turbines of this rating, presents an overall picture. When one examines in detail the efficiencies throughout the

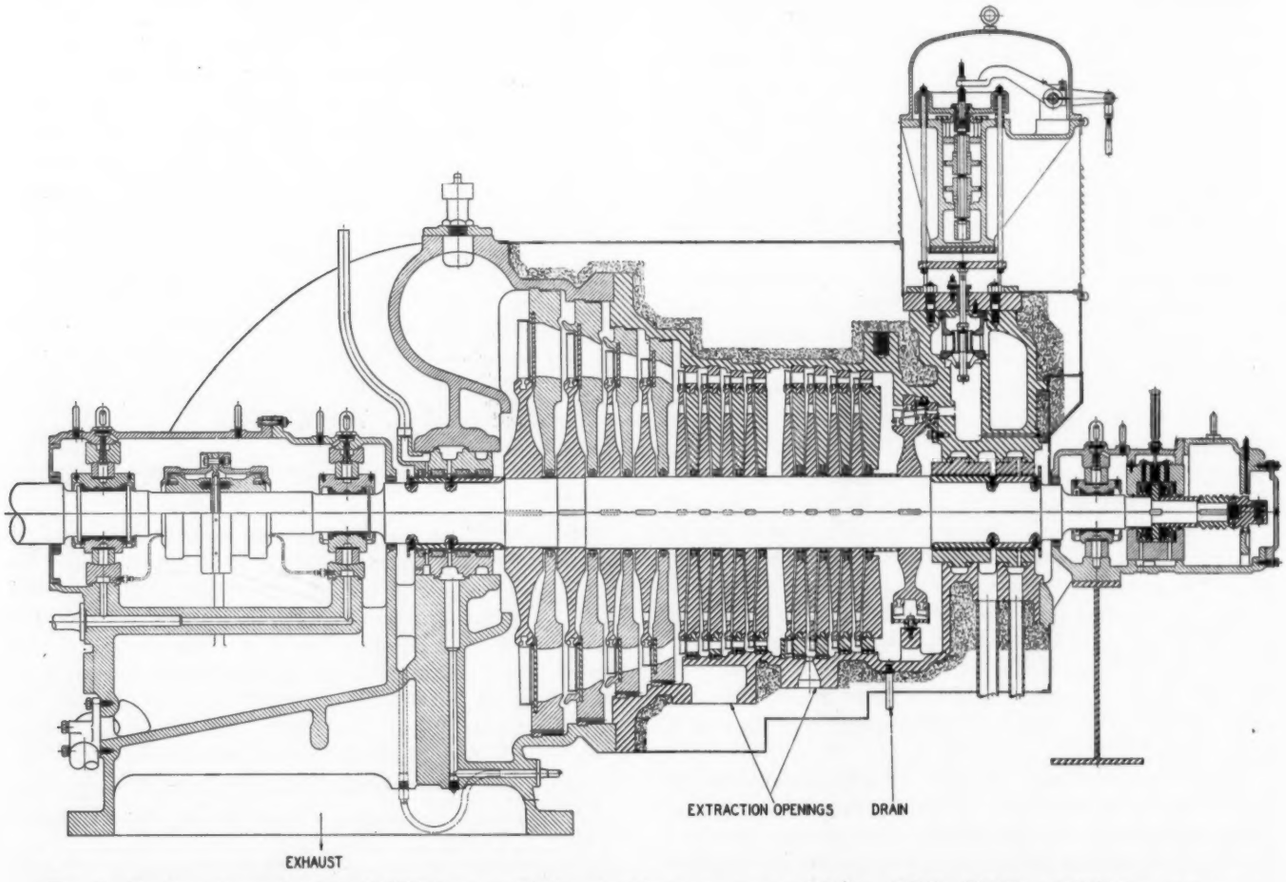


Fig. 5—Section through typical 4000-kw condensing turbine; steam conditions, 400 lb, 700 F and 28½ in. vacuum

machine, it appears that they are low in the high-pressure section largely as a result of short nozzles, leakage and rotation loss; they rise to a maximum in the intermediate section, and thence decrease in the exhaust end where the effect of the moisture and the leaving energy exert their influence. The conditions under which the last stage or two operate are particularly severe, not alone from the standpoint of moisture and supersaturation effects, but also as a result of the fact that the effective viscosity[5] of the steam at this point is greater than that of heavy lubricating oil.

Details of Typical Unit

The sectional view, Fig. 5, is sufficiently conventional to portray the general design features of most modern impulse-type machines of similar rating. The cylinder at the high-pressure end is a steel casting. This material is satisfactory for service up to 750 F but for operating temperature in excess of this, alloy cast steel is employed. In the exhaust end where the temperatures are low the cylinder is of gray iron. The two parts are joined near the atmospheric pressure zone. The steam chest which encloses the governing valves is confined to the upper half of the cylinder so that the first-stage nozzles can be effectively grouped for minimum loss and at the same time the bottom half is free for shielding the rotating wheel.

The nozzles and buckets throughout the machine are of stainless iron. In the high- and intermediate-pressure sections of the turbine the nozzle blocks are milled from bars to an effective profile. In the low-pressure end, the diaphragms are made from a profiled blade section, cast into an iron core. The blade is a 12 per cent chromium stainless iron with the addition of aluminum to inhibit air-hardening of the material as it cools from the casting temperature. The buckets are either milled or cold-drawn, depending upon their size, and are supported on forged disks, shrunk and keyed to the shaft. The nozzles and buckets in the larger low-pressure stages are designed with varying angles along their radial height to approximately satisfy the hydrodynamic condition of constant circulation.

The shaft packing of the radial type, located at the exhaust end and interstage on the diaphragms, is of leaded bronze. This is the material normally supplied for oil-less bearings. Rubbing tests under temperature on numerous materials, indicates that this is one of the most satisfactory applications for this service. Much of the labyrinth packing on the machine is of the full labyrinth type in which the restrictions are staggered on different diameters. By this means the leakage is reduced by about 40 per cent from that of a similar labyrinth having the seals all on the same diameter, and at the same time it has the operating advantage of confining any rubs that may occur on starting to the packing sleeves, instead of aggravating the distortion by allowing a rub directly on the shaft.

On the last stages of the machine, grooves are provided at the periphery of the diaphragms to assist in separating the moisture from the steam. The moisture problem is one of importance, for it not only accounts for a reduction in efficiency, but it introduces the equally serious maintenance problem from erosion on the blading and on the condenser tubes. While it is recognized that erosion is associated with three characteristics, the amount of moisture, the relative velocity of the drops in respect to the bucket[6] and the bucket material, the efforts of turbine designers to rationalize the situation have as yet remained discouragingly vague. Rough determinations of the stresses that must be withstood upon impact of the drop on the bucket, made by Cook and Parsons,[7] show that they are nearly proportional to the tip speed of the bucket, and that with a peripheral velocity of 1000 ft per sec they approach 75,000 lb per sq in., which may explain to some extent the difficulty in handling the problem.

Shielding and Moisture Draining to Retard Erosion

Two methods have been used with some success in attempting to retard erosion; these are shielding and moisture drainage. Shielding, with stellite at the inlet edge of the bucket, is universally employed for tip velocities in excess of 1000 ft per sec except in those cases where the moisture content is abnormally low. With moisture catchers an effort is made to take advantage of the fact that the water drops are crowded toward the tip of the bucket by centrifugal force and are then discharged with a radial component of velocity directed toward the moisture groove. However, the path of the moisture drop depends on its size which, in turn, depends upon the relative velocity between the drop and the steam. The high steam velocities in the low-pressure blading result in a very small droplet size, one which, because of its decreased mass in respect to the surface drag, tends to follow the path of the steam rather than to be projected radially; hence the effectiveness of mechanical separation is not high. Researches by the Elliott Company indicate that for the usual velocities and steam conditions found in the low-pressure end of a machine the drops probably have a maximum diameter of about 0.0001 in., which is merely "mist" size, whereas our model tests indicate that to separate water effectively by a mechanical contrivance, relatively low steam velocities with drop sizes of larger magnitude are necessary. These conclusions seem reasonable when it is recalled that tests on the effectiveness of moisture catchers on large turbines[8] indicate that much more drainage can be secured in the cross-over between the high- and low-pressure cylinder, where the velocities are low, than between the active stages.

Tests conducted on the drainage grooves of the type shown in Fig. 5 indicate that under favorable conditions between 6 and 10 per cent of the moisture present can be removed in the first drain groove, and somewhat less in the second. It is possible that some improvement could be shown if it were feasible to design the buckets without shrouds, but mechanical considerations, particularly from the standpoint of resonance, do not encourage this attempt.

While the effectiveness of separation by moisture catchers is not great, it is at the same time so inexpensive a scheme that it has been adopted as a measure of good practice in reducing erosion difficulties on turbines of reasonable size.

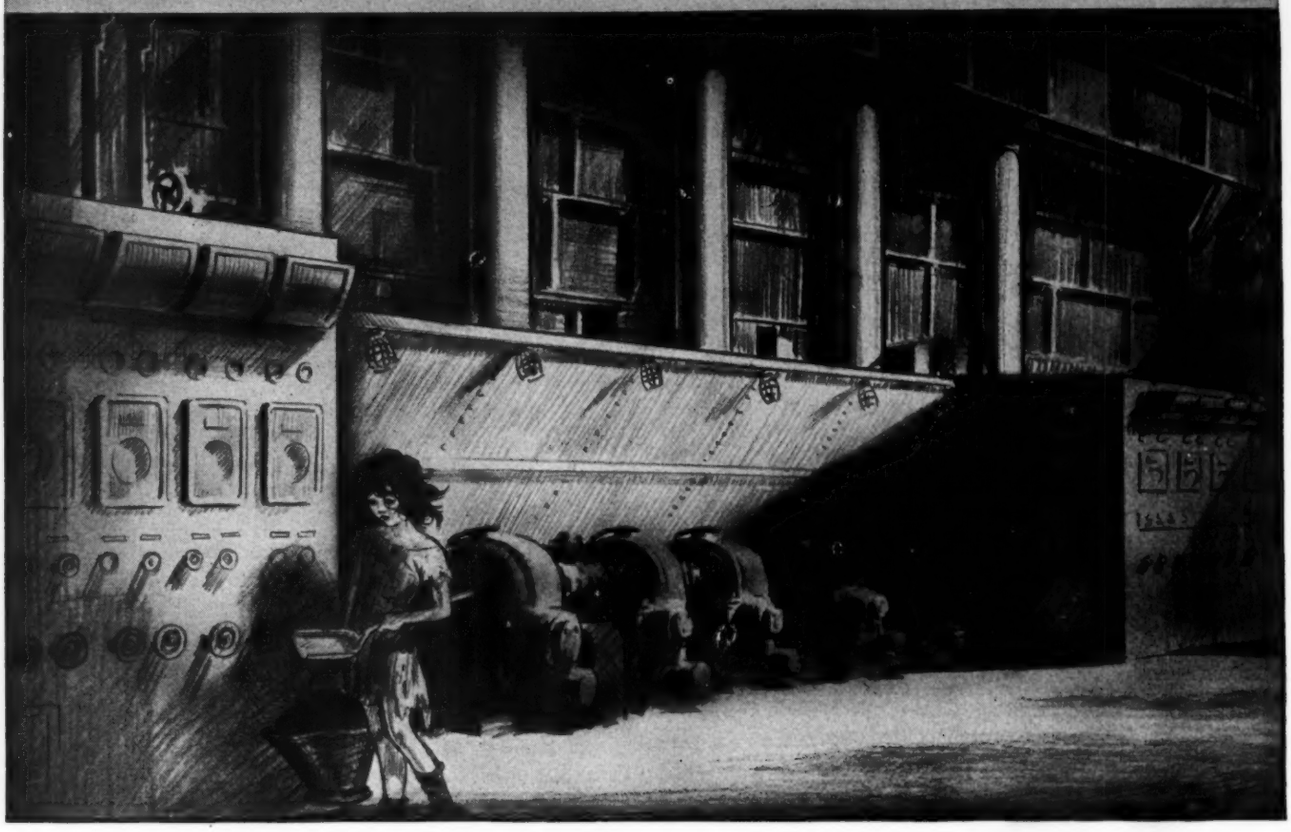
[5] By this term, reference is made to the kinematic viscosity of the steam.

[6] Practically this is represented by the tip speed of the rotating row.

[7] "Erosion by Water Hammer," Proc. of Royal Society, London, Vol. 119.

[8] Publication A3, "Turbines," Edison Electric Institute, New York, N. Y. 1933.

DON'T LET COAL BE THE PLANT PROBLEM CHILD



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GENERAL COAL

SPREADER STOKERS—

Their Application and Operation

Fig. 1—Three-unit spreader stoker installation

The following comments on mechanical spreader stokers cover the wide range of coals to which they are adapted, boiler capacities for which they are suitable, the most desirable combustion rates, setting arrangements and operating suggestions for cleaning fires and the proper control of fly-ash discharge. A typical unit is illustrated.

ADAPTED to a wide variety of coals and possessing the advantages of simplicity, low first cost, low maintenance and ready response to load variations, the spreader stoker has had a remarkable growth during the last few years. While the greater part of the installations have been in the Middle West, others have lately been made in the East and in the South.

With mechanical spreader-stoker firing, the coal is fed from a hopper into the path of rapidly revolving paddles which throw the coal into the furnace where the fines are burned in suspension and the coarser particles on the grate. Although this concept of firing dates back many years, the principles, while fundamentally sound, were not correctly applied in the early machines; hence it lagged behind other contemporary developments in fuel burning, particularly underfeed firing. Within recent years, however, experience with pulverized coal firing, together with intensive studies of furnace design, rates of heat liberation, turbulence, gas flow, fly-ash recovery and the use of water walls have been applied to the development of the spreader stoker with the result that its standard of performance has been raised and it is now being applied successfully to a wide range of fuels.

Spreader stokers can be operated at slightly higher combustion rates per square foot of grate than single-retort underfeed stokers, and the effective grate area is the projected area from wall to wall in both directions. The maximum continuous combustion rate for best results is about 40 lb per sq ft with good coal although as high as 55 lb may be carried for short peak loads. If too high a combustion rate be attempted, the high forced-draft pressure necessary and the resulting air velocity through the grates are likely to lift part of the burning fuel off the grates and result in excessive cinder carryover. Also high furnace maintenance may result.

The capacity limitation of spreader-stoker firing is not clearly defined. Steaming capacities from 4000 to 60,000 lb per hr represent a well established range, although in exceptional cases larger installations have been made.

Coals Adaptable

Particularly adapted to burning on the spreader-type stoker are midwestern bituminous coals from the Illinois, Indiana, Ohio, western Kentucky and adjacent fields. These are mostly high-volatile, high-ash, free-burning, clinkering coals, difficult to burn satisfactorily on underfeed stokers, although readily handled by stokers of the chain- or traveling-grate types. Sub-bituminous coals and lignite, which are mostly high moisture, free burning and clinkering, and which slack badly on drying, sometimes give trouble with underfeed stokers because of slow ignition, excessive siftings and excessive fly-ash carryover. They can be burned well on some stokers of the spreader type. This is because of the rapid drying of the fuel as it is thrown into the furnace; ignition is principally from the bottom, troublesome clinkering is unlikely, and due to the lower air pressure required fly-ash carryover can be controlled by careful operation. The same applies to certain bituminous and sub-bituminous coals of the Northwest. Eastern bituminous coals which are generally of the coking variety, with relatively low ash and high ash-fusion temperature, are usually burned on underfeed stokers or in pulverized form, but have also been found adaptable to some spreader stokers because the coking properties have little effect on the thin level fuel bed. Anthracite fines have also been burned on this type of stoker.

The popular sizing of coal appears to be $^3/_4$ in. and under, except in certain localities restricted as to fly-ash discharge, in which cases $^3/_4$ in. down to $^1/_4$ in. is generally

employed. However, sizing up to 1 in. has given satisfactory results in many cases.

Fuel-Bed Conditions

With most coals the active fuel bed on a spreader stoker grate consists of $^1/_2$ to $^3/_4$ in. of burning coal on top of a bed of ash. A thickness of one inch or more is likely to produce smoke when burning bituminous coal or lignite and closer regulation of the coal-air ratio then becomes necessary. The relatively small air space in the grate

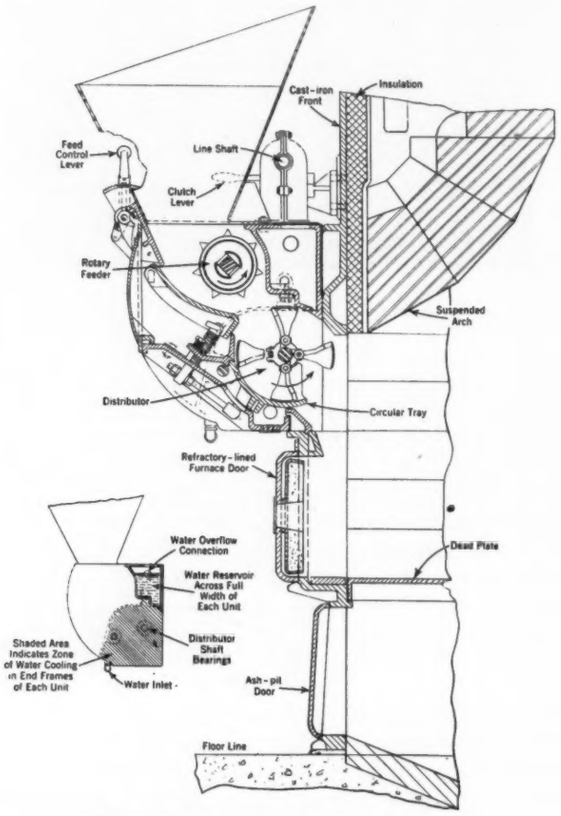

Fig. 2—Section through feeder unit of typical spreader stoker

is designed to create a constant resistance sufficiently large to minimize the effect of the varied resistance set up by the increasing thickness of the bed of ash. In this way uniform distribution and high velocity of the primary air are obtained through all portions of the grate surface so as to assure complete and rapid combustion of the thin fuel bed. The smaller coal particles, activated by the forced draft and "dancing" on the fuel bed, assist materially in maintaining its level.

Coking characteristics are scarcely apparent on such a fuel bed, as the thin, hot fire and rapid volatilization usually drive off the tarry matter before it can form large masses of coke. While individual particles may coke and swell, the fuel bed as a whole remains porous and free burning. Moreover, the underlying bed of ash is cooled by the entering air and is not agitated; hence, under proper operating conditions, troublesome clinker may be avoided. If, on the other hand, the bed of ash is allowed to build up excessively so that uneven air flow results, hand leveling with consequent agitation and mixing of coal and ash will be necessary and clinkers may form. Experience indicates that with most coals the fires should be cleaned when the ash bed reaches a thickness of about 4 in.

Cleaning Fires

When stationary grates are used the fires must be cleaned by hoeing out the ashes through the furnace doors. This is not only a hot and dirty task, but during this operation the efficiency and steaming capacity of the boiler will be affected; also furnace maintenance, where refractories are employed, is likely to be increased by the cool air admitted through the furnace door. However, this can be obviated by providing a dumping type of grate. In the latter case the loose granular ash is dropped into a flat pit and blown to the front by steam jets and readily removed.

By sectionalizing the stoker grates, either laterally for wide furnaces (with one section to each feeder unit), or from front to rear for deep furnaces (with two sections to each feeder unit, or in both directions for large stokers), the fires can be cleaned one section at a time. The procedure is, first, to shut off the feed to one unit; second, allow the fuel bed on that section to burn down; then shut off the air to that section and dump the ash. After doing this the coal feed is started again and a thin bed of green coal is built up before opening the forced-draft damper. Very little time is required and complete ignition is soon reestablished from the radiant heat of the adjacent active fuel bed. In general, for an average size boiler, fired by three feeder units and cleaned one section at a time, a total of fifteen to twenty minutes is required from the time the feed is shut off on the first unit until the fire is again burning actively on the third grate.

Factors that govern the timing interval between required cleanings are the percentage of ash in the coal, the combustion rate required to carry the load, and the opportunity as offered by load requirements; for it is obviously impossible to operate the stoker at its maximum capacity while cleaning fires. With small stokers this is likely to become a major factor which may be met by scheduling cleaning periods during lunch periods or between factory shifts. However, as the number of spreader units per boiler becomes greater this factor becomes less significant.

Response to Load Variations

A properly designed spreader stoker, under automatic control, will follow load variations very closely. In this respect it is somewhat similar to pulverized coal firing. When the load drops and the coal feed and air flow are reduced, there is so little green coal on the grates that there is no tendency for the steam pressure to overrun, as is the case with underfeed stokers because of the thick fuel beds. Likewise, when the load picks up and the coal feed and air flow are increased the heat input to the boiler responds almost instantly. Where loads are fairly steady hand regulation may suffice but for fluctuating loads, or where sustained high efficiency is sought, automatic control is desirable. In this connection a variable-feed method of regulation is preferable to the start-stop type.

During prolonged periods of light load where the demand drops nearly to zero for fifteen minutes or more, there may be a tendency for the fuel bed to burn out in spots, so that ignition is delayed when the load picks up again. In extreme cases of this kind a time control,

connected in parallel with the standard automatic control, may be employed to start the feeder or to increase the feed periodically, keeping enough fuel on the grate to maintain a good fuel bed, yet not enough to develop excessive steam pressure. It is well to point out, however, that there are certain types of load for which a spreader stoker is not suited, such as a heating load where the boiler is to be operated at very light loads for long periods, as during the early fall and late spring.

For overnight banking, a reasonably thick fuel bed is built up on the front of the grate and the dampers are set to allow a minimum natural draft. This is accomplished by closing tightly the stoker dampers and the overfire air dampers and "cracking" the boiler exit damper just enough to permit the escape of any gas that may be generated during the banking period. Some plants prefer to let the fire burn out, close all dampers to retain as much of the heat as possible and then rekindle the fire in the morning. This usually requires fifteen to twenty minutes before the boiler is ready to be put back.

Control of Fly-Ash Discharge

Because of the principle upon which the spreader stoker operates more fly ash and fine cinders are likely to be carried over by the combustion gases than with other types of stokers. This is especially true during peak loads or where high rates of combustion per square foot of grate surface are employed. However, the carryover can be minimized by proper design of stoker, adequate furnace volume and grate area, intelligent operation, proper draft control and provision for trapping and removing the fly ash.

In locations where excessive stack discharge may be objectionable, suitable settling chambers and means for ash removal should be provided. In settings with horizontal gas flow over a bridgewall, as with a horizontal return-tubular boiler, most of the heavier particles will drop out in the rear combustion chamber; whereas with settings of the vertical-pass types, such as with bent-tube or cross-baffled straight-tube boilers, the spaces ahead of and behind the mud drum and below the rear passes form natural settling chambers. Whatever fly ash leaves the boiler should be collected in settling chambers in the breechings or at the base of the stack. Vertical baffles to change or reverse the direction of gas flow are effective in diverting much of this material to the settling chambers without excessive increase in draft loss. For this purpose it is well to provide ample cleanout doors at the base of the stack.

For quick and easy removal of fly-ash deposits steam-jet vacuum systems are simple and relatively inexpensive, or fan systems can be used where high-pressure steam is not available. Also, mechanical conveyors of the screw or scraper type may be suitable.

In general, it is desirable that the fly ash, when collected, be discharged directly to the stoker ash pits or ash removal system, as the possible recoverable loss is seldom sufficient to warrant the ash being returned to the furnace for reburning. Moreover, return of the fly ash to the furnace has the disadvantage of increasing the carryover and furnace conditions are likely to be disturbed with resultant decrease in efficiency.

Excessive smoke has been associated with some spreader-stoker installations, but with correct furnace proportions, the use of air or steam jets and proper control of fuel and air, no greater amount of smoke should be emitted from the stack than in the case of other methods of firing.

There are a number of mechanical spreader stokers on the market, similar in principle but differing in details and refinements. One of these is illustrated in Figs. 2 and 3. In this design there is a cast-iron front on which is mounted a drive motor, a speed-reducing mechanism and one or more feeding units, each comprising a welded-steel coal hopper, a rotary feeder and a distributor. The rotary feeder acts as a crusher to break up large lumps of coal and any foreign material too large to pass the feeding mechanism may be easily removed by lowering the hinged feeder plate located at the front. A shear key on each unit protects the drive mechanism in case an obstruction lodges in the feeder.

The grate is of the dumping type, in the design shown, and is operated either manually or mechanically by steam or water pressure. Combustion air from the forced-draft fan is admitted under the grate into a plenum chamber divided into zones.

The distributor is located below the coal feeder and consists of cast-steel blades mounted in four rows, 90 deg apart, and arranged so that two opposing rows have the blades set at an angle to throw the coal to the right while the other two rows throw it to the left. The resulting crisscrossing of the fuel streams gives a consistently uniform distribution of fuel on the grates. Rotation is counter-clockwise, as viewed in Fig. 2, thus making the distribution "under-throw."

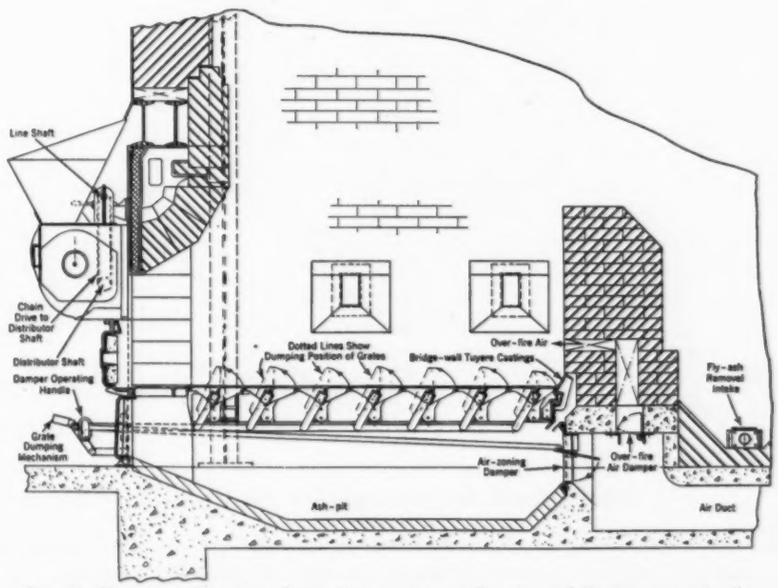

Fig. 3—Section showing dumping grates, ash pit and damper controls

In order to reduce maintenance water cooling is applied to the roller bearings of the distribution shaft.

With such a stoker the units are made up in several standard widths which can be applied singly or in multiples to meet the capacity required.

ADAPTABILITY

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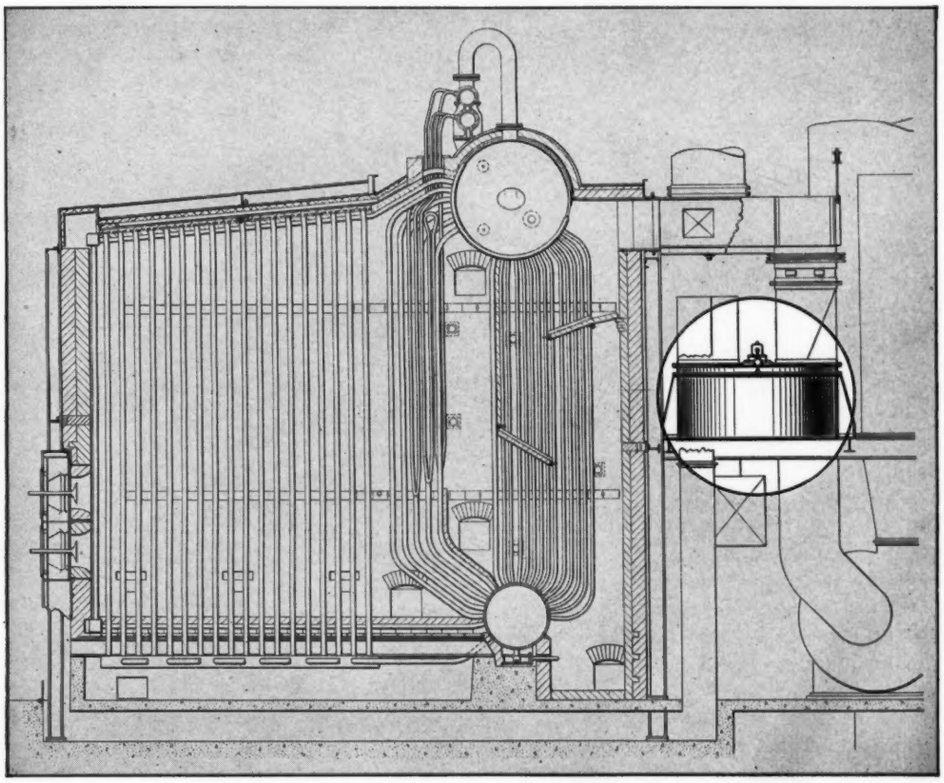

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SLAG DEPOSITS on Heat-Absorbing Surfaces and Means for Removal

By PAUL W. KEPPLER

Consolidated Edison Co. of New York, Inc.

The several factors that affect the form of ash accumulations on heating surfaces are discussed, as well as equilibrium conditions that produce self-cleaning. Removal by soot blowers, air, steam and water lancing is covered at length and precautions are cited, in the case of water, in order to prevent possible damage from undue chilling effects and strains.

SLAG formation seems to be governed largely by the conditions under which the suspended ash lands on the heat-absorbing surfaces. The density of the fly ash in the gas stream determines the quantity of the deposits for a given set of landing conditions. But even where this density is low, as is often the case with stokers, it is still high enough to cause serious trouble if the conditions are favorable for slag formation.

Effect of Ash Temperature

Most important among these conditions is that of ash temperature. In the first pass or through the central portion of a furnace, the temperatures of the ash and the gas seem to be closely related, although there is probably some lag in the temperature of the ash particles compared to that of the accompanying gas. For example, the conventional slag screen, consisting of widely spaced bottom tubes of the first pass, has been observed as failing to cool the suspended ash where the flame travel was insufficient between the screen and the next row of tubes. The ash particles in this case probably did not have sufficient time to follow the temperature drop of the surrounding gas, for the deposits were practically the same on the screen and on the next higher row of tubes. But where ample flame travel was provided, a very marked change in the deposits was observed.

The effect of temperature is apparently governed by the characteristics of the ash. If it arrives at the heat-absorbing surfaces above its fusion temperature, it seems to fuse while landing. Then if the gas temperature is very high, and especially if the gas velocities are also high, the resulting slag is likely to run freely.

At somewhat lower gas temperatures and lower velocities, ash, though arriving in the molten state and again fusing into a uniform layer, is likely to freeze on a water wall and form a rigid sheet. As this layer becomes thicker the surface exposed to the furnace becomes hotter, not only because it is itself becoming insulated from the water wall, but also because the gases likewise become insulated. Therefore, this slag will run eventually, at least, on the furnace side.

On the other hand, if the ash arrives at near its initial deformation or softening temperature, it cannot fuse properly when landing and a porous structure results. This may fall due to its own weight because it is not firmly bonded. Again, due to a gradual rise in the gas temperature resulting from impaired radiant heat transfer, this ash may eventually also become molten.

Finally, if the ash arrives below its softening temperature, no structure seems to be formed, and the resulting dust generally settles out by gravity on horizontal or slightly inclined surfaces. On tubes it builds up until reaching its angle of repose and on baffles until dead pockets are filled. These deposits appear to reach equilibrium conditions in a comparatively short time and retain them indefinitely unless disturbed by other changes such as the gas temperatures rising above the ash-softening temperature.

Where cleaning methods keep down the deposits, the transformation of the deposits and the rise in gas temperatures may be prevented.

Regardless of changing factors affecting the ash *after* being deposited, *during* its arrival on the heat-absorbing surfaces it seems to be governed largely by its own properties, fusing if fluid, building up loosely if near frozen, and forming dust if frozen.

Effect of Ash Velocity

Where the gas velocities are very low and where the ash settles out in consequence, as along some furnace walls, it is likely to be cooled by radiation while approaching a water wall. It may freeze into dry dust and not even stick. This probably explains why water walls often stay quite clean in spite of high temperatures not far from them.

On the other hand, where velocities of approach are high, as over certain convection surfaces and water walls swept by turbulent flow, time is probably lacking to

permit this chilling effect to take place. Moreover, the ash, instead of accompanying the gas past the surface or settling out loosely, is thrown firmly against the surface. This may explain why surfaces near burners often foul at a comparatively fast rate.

For a similar reason coarse grinding is reputed to aggravate slag deposits, despite the probability that coarse ash drops to the bottom more readily than fine ash.

Effect of Temperature of Heat-Absorbing Surface

Especially in the zone of dry porous deposits, the cooler the surface the less appear to be the deposits. For example, where water walls are composed of bare tubes with extended surface, the latter seems to collect ash at a much faster rate than the tubes, the difference probably being due to the higher temperature of the extended surface.

This relationship seems to be reversed with running deposits. Where enough high temperature heat reaches the surface, running slag tends to be very thin on a hot surface. But even on cool surfaces running deposits usually remain quite thin; much thinner than porous deposits.

Smoothness of the surfaces is probably also an advantage. Many engineers wash bare water walls, during outages, with water or with steam admitted to the cold furnace. This renders such surfaces perfectly clean and smooth. The thin ash scale normally remaining after thorough blowing with compressed air is said to accelerate deposits. But even with this residual slight scale, deposits appear to form at a slower rate before a uniform crust has developed.

Losses Due to Slag

Slag deposits, except possibly the very liquid type, impede heat transfer and affect draft loss. Furthermore, undesirable variations in gas velocity and in the rate of heat absorption can be caused by slag deposits on convection surfaces because the deposits tend to build up unevenly. Boiler tubes do not seem to be much affected by local increase in heat absorption, for, as a rule, circulation is ample and is likely to increase locally with increased heat absorption. But superheater tubes can be severely warped and bent, or in extreme cases may fail due to localized flame resulting from unequalized gas flow. At the same time gas temperatures may be excessive, largely because of the poorer preceding heat absorption. This also applies to superheater hangers, baffles, etc.

Dry dust deposits, on the other hand, do not appear to be particularly harmful in modern practice, except perhaps in the case of economizers or air preheaters if moisture be present due to the local temperature of the gases falling below the dew-point. Soot deposits forming a layer around the tubes are rare with modern coal firing. Stoker ash powders forming similar structures were observed at former low rates of driving, but seem to have been dispelled by present high gas velocities.

Self-Cleaning Surfaces

Many furnace walls are self-cleaning. The equilibrium conditions already discussed establish themselves and old deposits melt off or fall off at the same rate as new deposits are being formed. In many cases changes in output cause deposits to drop off. The shrinkage strains due to temperature variations are sufficient to break slag sheets or to loosen porous structures. Changes in output as low as 10 per cent are said to be sufficient for this effect in some cases. The writer has witnessed cases where no falling off took place until the unit was shut down, but then about 70 per cent of the slag sheets that had covered most of the water walls fell down in a few minutes. After that not much more fell off of its own accord but a little more could readily be pushed off. Self-cleaning by complete shutdown appears attractive at light load periods or when shutting down is desirable or necessary for reasons other than for cleaning. In most cases, however, it is probably an undesirable procedure to shut down for cleaning only, and considerable fouling is likely to be permitted before resort to this method of cleaning.

There are other cases where falling off of deposits is not very marked, but where some wall surfaces seem to "run" while others never acquire an appreciable coating.

Many furnace walls require no cleaning, yet the furnace outlet temperature seems to remain the same for a given condition of coal and output. If designed for this equilibrium condition, the furnace outlet temperature will always remain low enough. Bottom rows of boiler tubes seem to require some degree of cleaning in nearly all cases.

Behavior of Dust Deposits

Dust deposits, particularly those from pulverized coal fly ash, also seem to be self-limiting. They often streamline the passes and cause little or no draft loss. In some cases rough measurements even indicate a decrease in draft loss. The drop in heat transfer rate is slight. Where bypass dampers or other superheat controls are used, no loss can result from dust deposits on the superheater, for although the coefficient of heat transfer may drop slightly this is offset by changing the amount of gas bypassed. Experience indicates that it is safe to leave dust deposits on superheaters, providing the gas temperatures can be kept below the softening point of the ash. This, in turn, can easily be accomplished in many cases by keeping the surfaces ahead of the superheater reasonably clean. If there is danger of the gas temperature rising above the softening point in the superheater, then it is safer to remove the dust accumulations periodically. This may very readily be accomplished by means of soot blowers or by hand lancing.

The writer's experience in cleaning has been confined largely to convection surfaces, but it is probably applicable to furnaces as well. Dust and porous slag may easily be removed by compressed air or steam, hand lances or soot blowers, whereas dense slag requires chilling by water, either in a solid jet or mixed with air or steam.

Steam and Air Lances

These require a good jet to be effective. The nozzle should be carefully shaped by a blacksmith, as indicated in Fig. 1. For hot locations it should be large enough to cool the lance, yet small enough to build up pressure. Air lines should be large, 2 in. or more, to preserve the air pressure. A 1-in. hose causes only a small fraction of the pressure drop of a $^3/_4$-in. hose. Since drop through the hose may destroy much of the available pressure (50 per cent or more) a 1-in. hose is believed to be more

economical for many cleaning operations. The layout of doors and platforms is most important for economical lancing, as illustrated in Fig. 2. With the platform and doors as shown, very simple lances may economically clean this surface providing the gas temperature is below softening temperature. The arrangement of Fig. 3, (which has not been experimented with) is probably still more economical, but requires a special heatproof lance material or some cooling method, such as a small quantity of water added to the air.

Soot Blowers

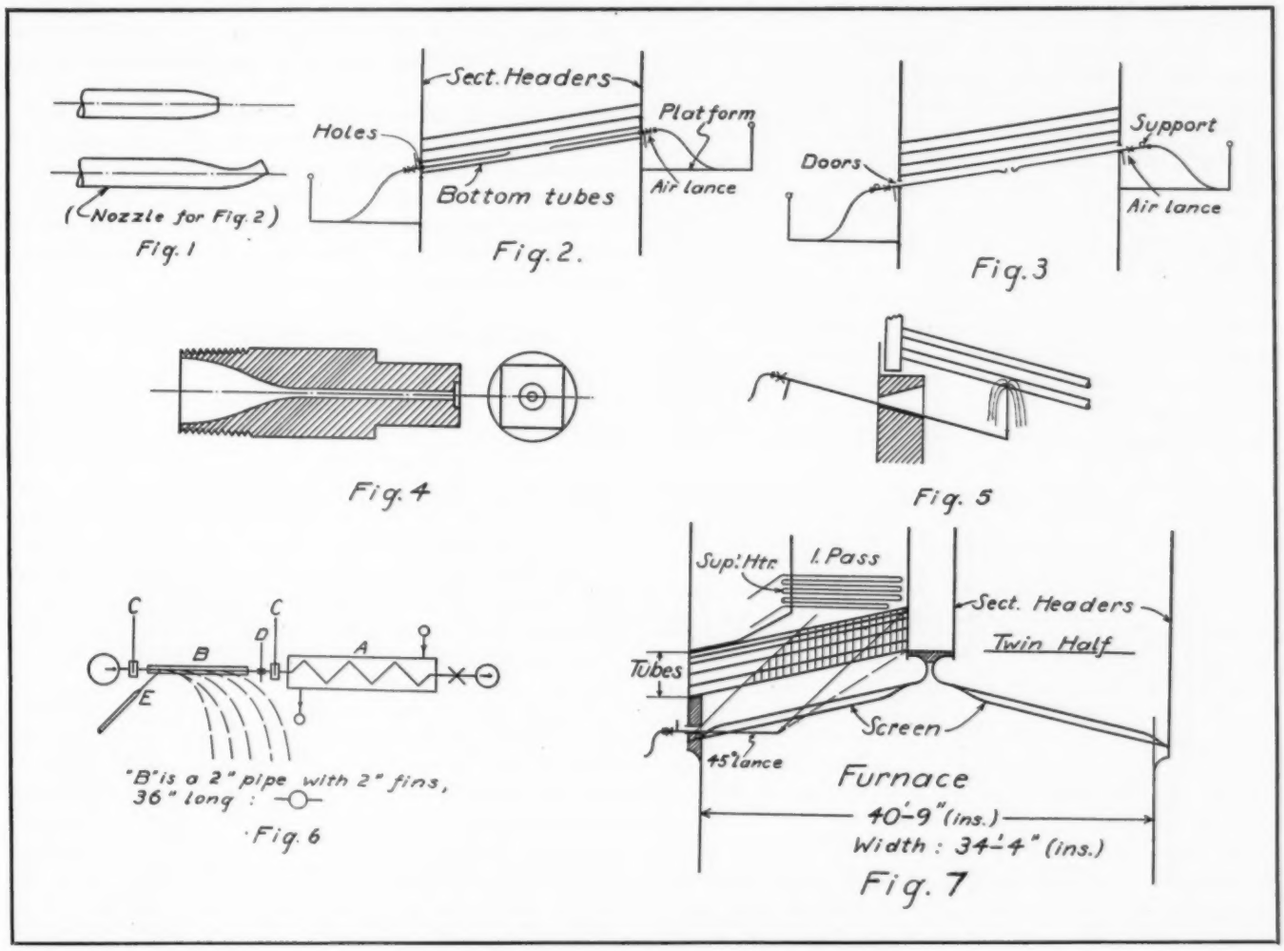
The conventional stationary or rotating soot-blower element has given excellent service where gas temperatures were sufficiently low, even for the removal of porous slag at the very entrance to the first pass. But, generally, warping and burning of the elements has been so bad that retractable soot blowers are today preferred for high temperatures. For removing dense slag, water has frequently been added to the steam.

In many instances it has been found necessary to employ higher steam pressures at the soot-blower head, necessitating larger supply pipes, where the steam is taken from the low-pressure header, or passing high-pressure steam through a reducing orifice. Pressures as high as 300 to 350 lb have been used in some cases and proved most effective.

Water Lances

Water lances exert a considerable chilling influence. Surfaces remain wet after the jet has passed away indicating that they have been cooled down to 212 F or even lower. Experience seems to prove, however, that surface strains are harmless when such lancing is properly applied and certain surfaces avoided.

One effect of water lancing is to shrink the tube longitudinally. This is apparently harmless with bent tubes, but with straight tubes, trouble in many cases has been experienced from rolled-joint leakage. The writer's experience, however, indicates that this may be avoided by proper care and control.



Figs. 1 to 7—Illustrating means of water lancing

Obviously, rolled joints themselves should be kept dry. Superheater tubes are likely to warp due to chilling and should be wetted as little as possible. Superheater hangers, brickwork and baffles may also be severely damaged by water. These should all be kept nearly dry.

While experience shows that where care is exercised no damage results from water lancing, the danger is nevertheless a very real one. Water lances should therefore not be applied without great caution. A very small jet should first be tried, possibly $^1/_{16}$ in., on a small portion of the surface. This surface should then be carefully inspected. If no damage can be detected, then a larger

jet, possibly $^3/_{32}$ in., appears safe, still on a restricted surface. Again careful inspection should be made at every opportunity. If several months' restricted use proves this $^3/_{32}$-in. jet safe, then it may be put into general use. But careful inspection should continue for about a year. Until it has been definitely established over a long period that $^3/_{32}$-in. jets are perfectly harmless, $^1/_8$-in. jets should not be used. These jet sizes assume 400 lb pump pressure. For other pressures they may be varied so that the product of the square of the diameter times the square root of the pressure is equal to the same product for the values here given.

Protection Against Tube Shrinkage

To protect straight tubes from shrinkage, the rate of chilling should be held below the rate of flow of steam heat in the tube at the point of chilling. In this way shrinkage is confined to the chilled surface, because the flow of water through the jet should be kept low. Effectiveness may be sustained by high water pressure and a solid jet, nearly free from spray. High pressure may be obtained in many ways. Reciprocating pumps have several important advantages. They do not overheat at zero flow; they also drop their pressure in case of hose failure and a light, convenient hose may therefore be used with safety; and 400 lb pressure may be obtained at low cost. A solid jet may be achieved as shown in Fig. 4 which shows a smooth entrance followed by a straight section. The tip is recessed to protect it from damage. Use of stainless steel is desirable to reduce wear. Effectiveness may be obtained with a $^3/_{32}$-in. jet, although a $^1/_4$-in. jet is more powerful.

The heat of the steam in the tube should be kept high by running the boiler at a high output during straight tube lancing. This should be not less than two-thirds of full output.

A further safeguard consists of keeping the chilled surface short by applying the jet as nearly perpendicular to the tube as is feasible; see Fig. 5. Shrinkage is thus restricted to the short length of the chilled surface. Furthermore, the total chilling influence is obviously much less than that resulting from running the jet along the tube, because in the latter case the water splashes away, doing less tube scrubbing and distributing its cooling action between a number of tubes.

It would be desirable to know exactly how great this heat transfer from water jet to tube is, for then the maximum permissible chilling effect could be estimated, assuming that the steam heat in the tube can be calculated from a heat balance based on gas temperatures. The writer has made some rough tests as shown in Fig. 6. The surface and jetting arrangement shown only crudely resemble actual conditions. Therefore, the data gathered may be used only as an indication.

Ahead of the feedwater cooler *A*, Fig. 6, previously used for oxygen testing, was placed the surface *B* resembling a 4-in. furnace wall tube with extended surface; also the thermometer wells *C* and the $^1/_4$-in. mixing orifice *D*. A cold-water jet *E* was directed against the surface *B*. To get a greater value for heat transfer, the 90-deg angle of impingement ordinarily used, was replaced by a 45-deg angle. Only 50 lb pressure was available for jetting, but by holding the nozzle at a distance of only 6 in. from surface *B* at least as much impingement was produced as in the actual case. The fins used were unusually long but being $^3/_8$ in. thick they could readily conduct the small heat flow here encountered. Flows were determined by volumetric measurement before and after the temperature determination. Very thorough wetting by about the same flow as delivered by a $^3/_{32}$-in. jet at 400 lb pump pressure gave a heat transfer of only about 3 Btu per sec with a feedwater temperature of about 300 F. Due to higher temperatures of actual tube surfaces more than double this heat transfer actually takes place. However, as a rule, boiler tubes and many water-wall tubes receive heat from the furnace at high outputs at the rate of over 5 Btu per sec per ft of exposed tube length. Therefore, if several feet of tube length are left dry below the point of chilling, cooling of the tube is prevented by its own steam bubbles except at the point of impingement. The writer when jetting straight tubes has in every case left at least 5 ft of tube dry below the point of chilling, except in one case to be described later.

It is hoped that schools of engineering may make available accurate data on heat transfer of both solid jets and water and steam mixtures. This would make possible more powerful cleaning tools and the complete elimination of strains of any kind. In the meantime empirical methods are necessary, and these should be as careful as possible.

Abandoning Observation During Lancing

Still another precaution consists of abandoning observation while lancing because during observation the lance is usually left in one spot. This is not harmful under safe conditions, but there may be cases where adherence to all the foregoing rules is impossible, and where the thermal inertia capacity of the metal and water of the tube must be utilized to reduce shrinkage. By substituting for movement according to observation, a methodical oscillating motion, the chilling influence is uniformly distributed among a great many tubes, and obviously the chilling influence per tube is reduced very greatly.

An example of this is shown in Fig. 7. The section below the superheater has to be cleaned. Deposits are dense on the screen, but due to these tubes being bent, cleaning presents no problem. The deposits are porous on the next six rows of tubes. A water lance is effective, but space limitations restrict it to an effective length of 7 ft 6 in. By oscillating and advancing this lance, the shaded section may be lanced. This is sufficient, but obviously both the bottom of the superheater and many rolled joints get wet. The distance to be covered makes a powerful $^1/_8$-in. stream desirable. By employing a methodical oscillating motion damage to superheater and rolled joints has been prevented. Where oscillating motion is used, longitudinal advance of the lance should be made at the end of each half stroke to permit the lance to progress constantly on to new ground where it is most effective. Motion without observation is effective for porous, more or less uniformly distributed deposits. Concentrated dense deposits, that require individual chilling and removal, make observation necessary.

Cleaning of water walls by water lancing seems to have been satisfactory in some stations but in others it apparently has led to some trouble with the extended surface or with the refractory behind the wall. However, by careful analysis of the chilling action of the water and

by cautious experimentation on a small scale, it should be possible to develop consistently safe cleaning methods for many water walls.

Cost of Cleaning Operations

Most cleaning operations are costly. Air lances require large compressor capacity much of which is chargeable to lancing. A $^1/_2$-in. jet at 30 lb pressure consumes about 120 cu ft of free air per min and about 25 kw hr at 100 lb compressor pressure. A $^1/_2$-in. steam jet at 100 lb pressure and 600 F temperature requires about 1000 lb of steam per hour. Soot blowers usually take a large flow of steam but it should not be necessary to operate them for long periods. The quantity of steam used per day under operating conditions should be carefully specified and checked for any given cleaning requirement. Water lances require nearly 0.1 lb of coal per pound of water, so that a $^3/_{32}$-in. jet at 400 lb pump pressure wastes about 0.1 ton of coal per hour, and a $^1/_8$-in. jet nearly double that amount. The cost of labor for cleaning is likewise high, even for soot blowers, the maintenance and fixed charges of which are also appreciable in many cases.

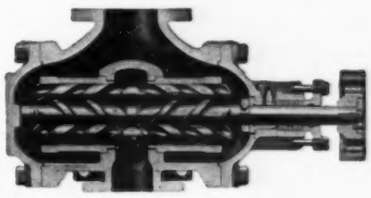
Auxiliary power, thermal losses, labor, fixed and maintenance costs should be carefully checked and held to a minimum. Besides selecting the most economical cleaning tools among soot blowers or lances, the most economical cleaning schedule must also be found. Slag should not be allowed to get far ahead of cleaning operations, but it is too costly in most cases to almost eliminate slag by very frequent cleaning. The best compromise should be found by careful experiments. More frequent cleaning than once per day is probably not justified except in extreme cases. There are also cases where gradual fouling should be permitted, followed by periodic outage, because cleaning may be very much cheaper during outage, accessibility is much better and slag is more brittle when cold.

Cleaning of all or of a large portion of the furnace walls is likely to be costly as these surfaces are so extensive that many platforms or soot blowers must be provided and much labor is required. Tubes at the entrance into the first pass, however, are sufficiently concentrated to make cleaning more efficient. Superheater tubes, on the other hand, are usually too tightly packed for the convenient dislodging of slag deposits, unless these are very porous. Dense slag is extremely difficult to remove while in service because water must not be used in this part of the unit. It is therefore a great advantage if the furnace and superheater are self-cleaning.

Where a superheater bypass damper is used it is likewise desirable to have enough boiler surface and flame travel below this damper to convert the ashes into dust even at maximum bypass flow. Otherwise, this damper and the surfaces immediately beyond it may require considerable cleaning.

While deslagging deserves much thought the best program is likely to result from reduction of harmful deposits by improved design. Probably all manufacturers have produced many self-cleaning furnace walls and superheaters and many entrance banks into the first pass that are very easy to keep clean. It is quite possible that these fine performances could be consistently reproduced at a saving in fixed and operating charges.

THE IMO test room has quantities of various oils and facilities for heating them to various temperatures, also dynamometers and other testing equipment, so that any IMO pump can be tested under the conditions for which it is sold, thereby assuring satisfactory operation in the user's plant.



Publication I-51 gives data on various IMO applications.

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Coal Facts

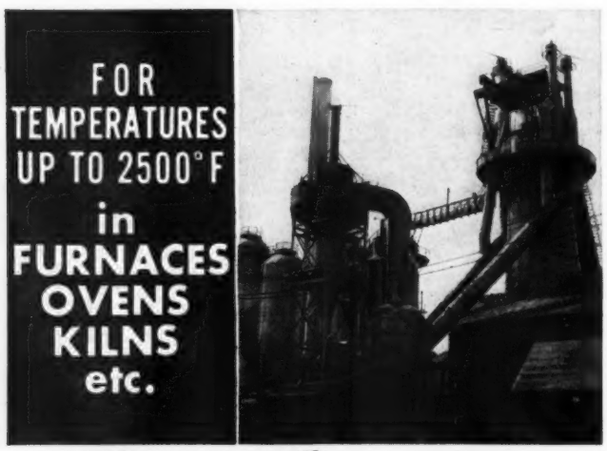
T. A. Day, of Appalachian Coals, Inc., speaking before the Rotary Club of Knoxville on January 9, cited some interesting facts about coal, which it is believed are not widely appreciated. He said, in part:

Coal actually tops the list in so far as employment of labor is concerned, with the exception of wage earners employed by the railroads. In the United States today there are more coal miners than steel workers, more than those engaged in the production of motor vehicles, and more than are employed in the cotton mills or shoe factories. The railroad total, according to the latest I. C. C. reports, is 975,625; and coal mining ranks second with 565,672, of which 462,403 are engaged in the production of bituminous coal and 103,269 in the anthracite industry.

Each ton of bituminous coal mined brings $1.27 to the miner in the form of wages; whereas the labor cost in the production of four barrels of fuel oil, the equivalent of one ton of coal, is 68 cents; that in the production of 20,000 cu ft of natural gas, also the equivalent of a ton of coal, is only 8 cents, and that of 2000 kwhr of hydroelectric energy is less than one cent. These figures point their own moral, that displacement of coal by substitute fuels is one of the major factors in the present unemployment problem.

The bituminous coal industry has fostered and promoted the use of automatic stokers and other coal-burning devices. During the first nine months of 1939, the U. S. Department of Commerce reports the total sales of stokers as 71,661, an increase of 5797 over the same period of 1938. Of this total 54,526 stokers were of domestic size.

Time, effort and money are being spent in both pure and commercial research activities and the possibilities of finding more uses for coal have received a great deal of attention. In this connection a plant capable of producing 4 million pounds of Nylon yarn has just been opened and already employs 850. Nylon is a coal product. Other products also derived from air, water and coal gas are "zerone," an anti-freeze mixture, and ammonia.



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Midwest Power Conference Scheduled for April

The Annual Midwest Power Conference will be held this year on April 9 and 10 at the Palmer House, Chicago. As in former years, it is being sponsored by Armour Institute of Technology in cooperation with a number of engineering colleges and local sections of the engineering societies. The tentative program to date is as follows:

TUESDAY, April 9

9:00 a.m.
Registration

10:00 a.m.
Opening Meeting
Address of Welcome by Loran D. Gayton, City Engineer, Chicago, followed by a response for the cooperating institutions

10:30 a.m.
Power Fallacies by Philip W. Swain, Editor of *Power*

11:15 a.m.
Gas Turbines

12:15 p.m.
Joint luncheon with A. S. M. E.

2:00 p.m.

SMALL POWER PLANTS
Diesel Units
Diesel vs. Steam Units in Competitive Fields
Maintaining the Optimum in Steam Generator Efficiency by Parker A. Moe, Superintendent of Power, Milwaukee Works, International Harvester Co.

3:45 p.m.

ELECTRICAL TRANSMISSION
Protection of High Voltage Lines by Philip Sporn, Vice President, American Gas and Electric Service Corp.
Bus Arrangements and Switching
Electrical Distribution

POWER-PROCESS
Paper Mill Power by Grover Keeth, Marathon Paper Mills Co.
Power for the Refinery by R. L. Meyer, Superintendent of Utilities, Whiting Refinery, Standard Oil Company of Indiana
Power for the Chemical Plant

6:45 p.m.
"ALL ENGINEERS" DINNER, Red Lacquer Room, Palmer House

WEDNESDAY, April 10

9:30 a.m.

FUEL PROBLEMS OF POWER PLANTS
Stratification of Gases in Coal-Fired Furnaces by John M. Drabelle, Iowa Electric Light and Power Company.
Pulverized Coal, by Martin Frisch, Chief Engineer, Boiler and Pulverizer Division, Foster Wheeler Corp.
The Intermittent Burning of Gas and Pulverized Coal
The Gas-Fired Plant and Its Problems by E. L. Tindall, Superintendent, Fuel and Combustion, Carnegie-Illinois Steel Corp.

HYDRO POWER
Small and Medium Sized Hydro Plants
The Santee-Cooper Project by L. F. Harza, Chicago
The Hydrological Factors in the Design of a Dam, by James S. Bowman, Tennessee Valley Authority

12:00 Noon
Joint Luncheon with A. I. E. E.; Speaker, L. W. W. Morrow, McGraw-Hill Publishing Co., Chicago

1:00 p.m.
Inspection Trip

STEAM ENGINEERING ABROAD

As reported in the foreign technical press

Tests on Combustion Turbine

The results of load tests on a 4000-kw combustion-turbine set are reported by Prof. A. Stodola in *Engineering* of January 5, 1940. The unit, built by Brown Boveri & Co. of Baden, Switzerland, for a bombproof power plant, consists only of the essential components, such as compressor, combustion chamber, a turbine exhausting directly to atmosphere and the generator; a heat exchanger between the exhaust gases and the combustion

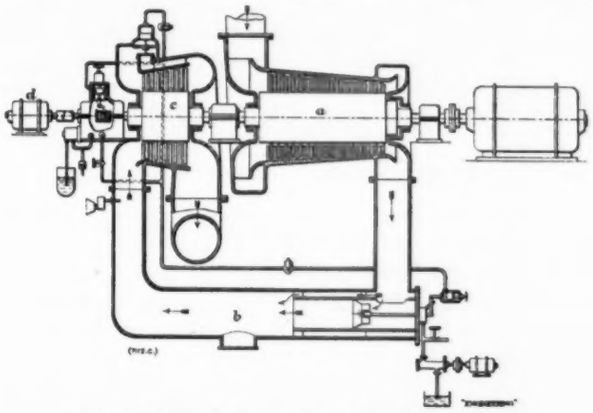

Fig. 1—Section through combustion turbine

air having been omitted because of the standby nature of the service. The set runs at 3000 rpm. Despite the simplicity of the arrangement, the tests showed an efficiency, referred to the heat contained in the fuel and the heat equivalent of the electrical output, of 17.38 per cent. A feature worthy of note is that the plant requires no cooling water.

Referring to the cross-section, Fig. 1, *a* is the compressor, which compresses the air to 40–60 lb abs; *b* is the combustion chamber; *c* is the turbine with multistage reaction blading; and *d* is the starting motor. Only part of the total air is used for combustion, so that the temperature reaches about 2500 F, the remainder subsequently mixing with the gases after combustion to bring the temperature down to about 1000 F.

	Test No. II.	Test No. III.
Fuel consumption .. .. lb. per hour	4,338	3,649
Temperature at the gas-turbine inlet deg. F.	1,025	916
Net output at terminals .. ... kW	4,021	3,057
Net output at coupling .. .. ,,	4,184·3	3,193·8
Consumption per kWh at terminals lb. per kWh	1·078	1·193
Net calorific value of fuel B.Th.U. per lb.	18,257	18,257
Heat supplied by fuel per kWh at terminals B.Th.U.	19,680	21,780
Thermal efficiency referred to output at terminals .. .. .. per cent.	17·38	15·67
Efficiency referred to output at coupling ,,	18·04	16·37
Or, for the guaranteed conditions, namely: air temperature 68 deg. F.; speed, 3,000 r.p.m., the figures are :—		
Output at terminals .. .. .. kW	4,000	3,026
Output at coupling .. .. .. ,,	4,163	3,162
Temperature at gas-turbine inlet deg. F.	998	890
Thermal efficiency referred to terminal output .. .. .. per cent.	17·38	15·67
Thermal efficiency referred to coupling output .. .. .. per cent.	18·04	16·37
Fuel consumption, based on fuel having a net calorific value of 18,000 B.Th.U. per lb. .. .. lb. per kWh	1·000	1·210
Guaranteed consumption based on the net calorific value of 18,000 B.Th.U. per lb. lb. per kWh	1·164	1·260

The test results are shown in the accompanying table and Fig. 2 is a view of the assembled unit on the test floor.

Condenser Tubes

According to V. Walker, in the *Electrical Review* (London) of December 29, the present tendency is to replace condenser tubes of admiralty metal with those

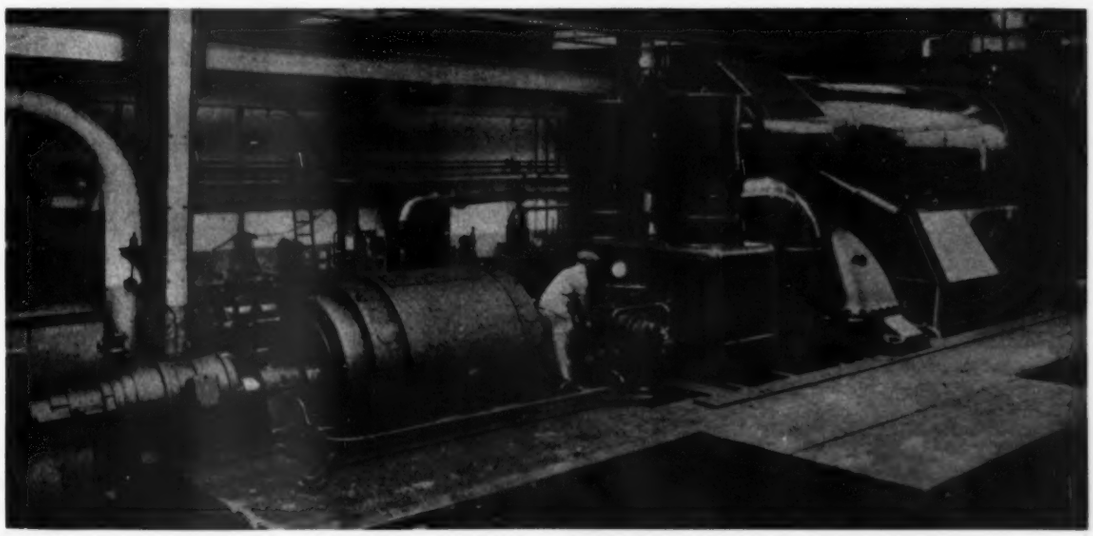

Fig. 2—Combustion-turbine set on testing floor

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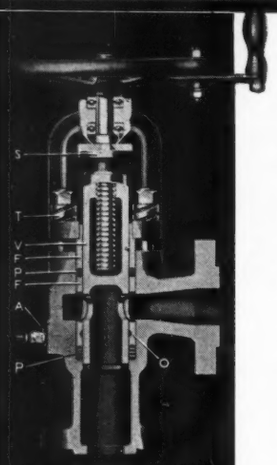

Yarway Seatless Blow-off Valve. Operation: After Valve is closed, shoulder S on plunger V contacts with upper follower gland F, forcing it down into body and compressing packing P above and below port. Annular groove O connects with Alemite fitting A for lubricating plunger and packing. Yoke springs T maintain continuous pressure through follower gland F on packing rings P.

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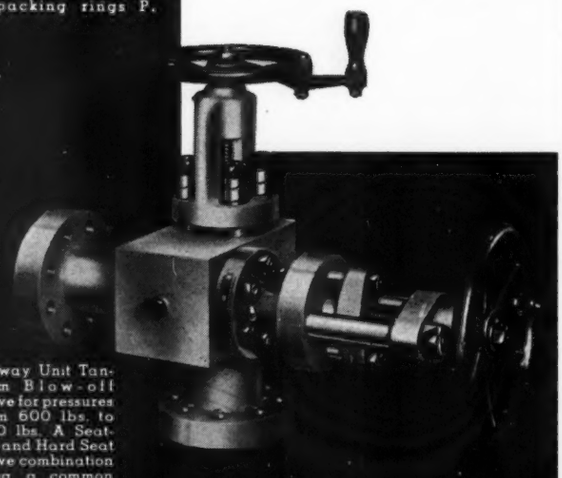


Yarway Unit Tandem Blow-off Valve for pressures from 600 lbs. to 1500 lbs. A Seatless and Hard Seat Valve combination using a common forged steel body.

YARWAY

BLOW-OFF VALVES

of more corrosion-resistant alloys, such as aluminum-brass or cupro-nickel, both of which also offer better resistance to cavitation.

Cupro-nickel alloys in common use abroad for condenser tubes contain 20 to 30 per cent nickel and the balance copper. Sometimes 5 per cent zinc is added to the 20 per cent alloy, but this has little bearing on the corrosive properties. This alloy is employed where all-round corrosion resistance is desired and has given excellent results with contaminated water supplies, as well as with salt and brackish waters. It has good physical properties at both normal and elevated temperatures and is finding increased use in marine condensers.

Aluminum-brass contains 76 per cent copper, 2 per cent aluminum and 22 per cent zinc. It has shown superior service where the circulating-water velocities are high or where turbulence tends to produce end corrosion. It also has the property of forming on its surface a thin tenacious film which makes the tube highly resistant to erosion as well as corrosion. However, aluminum-brass is not resistant to dezincification; but a modification of this alloy, containing 82 per cent copper, 2 per cent aluminum, 1 per cent tin and 15 per cent zinc has been found to resist dezincification satisfactorily.

Testing Pipe Welds

In *Wärmewirtschaft* for November 1939, Dr.-Ing. K. Jurczyk reviews the difficulties encountered in testing welded pipe lines sectionally with water, air or gas, due to the large quantities of fluid required. He recommends a scheme for a simpler means of testing welded pipe joints, as shown in Fig. 1, applied to a butt-weld joint. A collar is pushed over the welded joint and is welded to the adjoining pipes by two light seams. This collar, which may have a thinner wall than the pipe, carries a threaded nipple to which a pump may be connected. The space between the collar and the pipe is then subjected to any water pressure desired for the test. If the

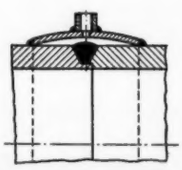


Fig. 1—Collar for testing weld

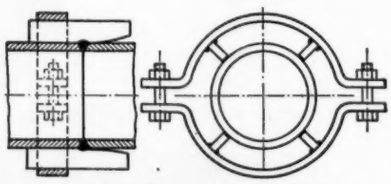


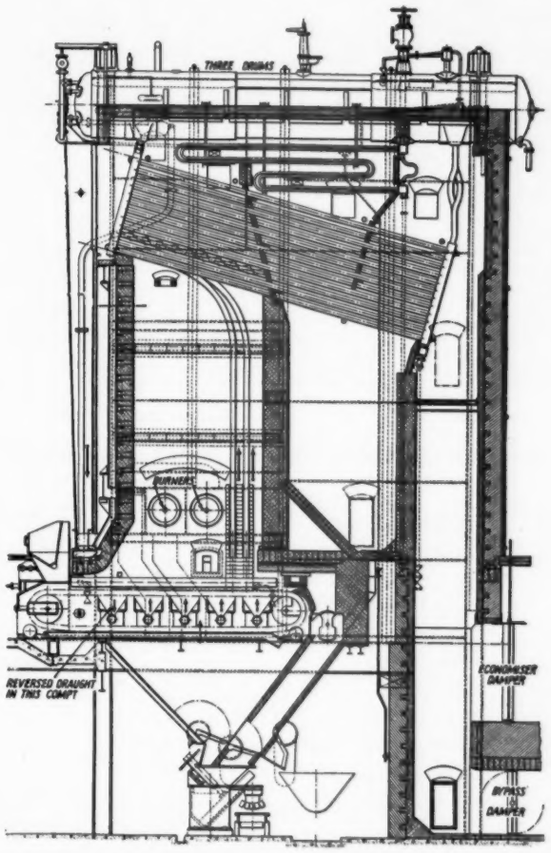
Fig. 2—Clamp for centering tube ends

pressure stands up for a predetermined time without having added any water it shows that the pipe weld is tight. The light weld seams of the collar may be easily inspected for tightness. After the test, the nipple is plugged and the collar serves as a reinforcement for the pipe weld. The author states that the collar test is cheaper than testing by filling the entire pipe.

The arrangement shown in Fig. 2 is suggested for ease in centering tube ends to be welded. It is made of two half guide clamps which are either joined by two bolts or by a bolt on one side and a hinged joint on the other. Into each half clamp are welded two guide pieces as shown. The clamps are placed around one of the pipes adjacent to its end and then clamped so that the guides bear hard against the pipe. After use on one joint the bolts are loosened and the guide clamp is pushed along the pipe to the next joint.

Burning Pitch and Coke Breeze

The setting of a 30,000-lb per hr boiler burning pitch in combination with coke breeze at a British gasworks is described in *The Power and Works Engineer* of January 1940. By eliminating the ignition arch and providing a separately supported front wall it was possible to employ a shorter chain-grate stoker than is customary with coke breeze. Reference to the arrows in the illustration shows that the front compartment of the stoker employs reversed draft. Side-wall water boxes are provided and



Furnace for burning pitch and coke breeze

the induced-draft fan has a grit collector. Because of the erosive nature of the coke breeze, the cast-iron oscillating chutes feeding the stoker hopper are internally coated with Linatex.

The pitch is fired through the side wall and is preheated to 480 to 500 F in a steam-jacketed pipe-line circuit, the temperature being maintained by a thermostat. The burners have quick-release devices so that a clean one can be readily substituted for one that has become clogged. Coke breeze alone is burned when bringing the boiler up from cold condition, the pitch not being turned on until the refractories have attained normal operating temperature.

Russian Mercury-Vapor Turbine

Under the title "Modern Turbine Technique and Design," in *Engineering and Boiler House Review* (London), D. W. Rudorff refers to a mercury-vapor turbine of 4000 kw capacity laid out in 1938 by the Soviet Central Boiler

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Write us for more information about this way to use applied fuel engineering. Or better yet, arrange to talk it over with us, either at your office or at ours. There is no obligation.

FUEL ENGINEERING COMPANY OF NEW YORK

FUEL AND POWER CONSULTANTS SINCE 1907

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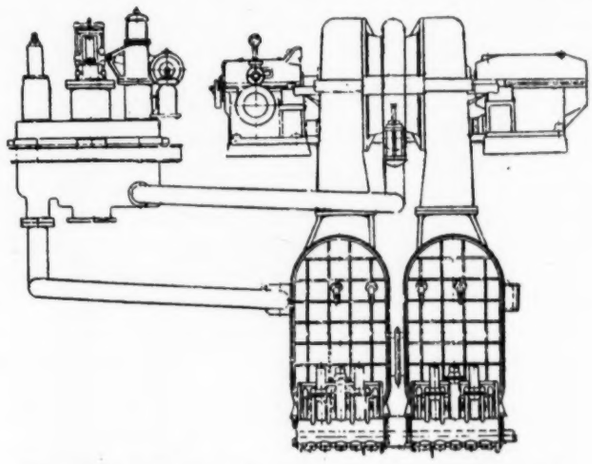
and Turbine Institute. This employs an initial mercury-vapor pressure of 142 lb per sq in. and 955.4 F temperature, the mercury vapor being exhausted to the condenser at 1.7 lb pressure and 494.6 F. The turbine rotative speed is 1500 rpm.

An interesting feature of this machine is the double-flow arrangement, which was chosen in order to eliminate superatmospheric pressure at the shaft glands and thus facilitate vapor tightness. Each half of the double-flow turbine exhaust has its individual condenser, the steam generating surfaces of which have forced circulation. Mercury pressure equalizing pipes are provided between the two condensers and the latter are located in the basement below the turbine with the flanges of the turbine exhaust connected directly to those of the condensers to minimize possible leakage. Two horizontally curved pipes connect the valve chest with the turbine inlet passage and two bypass pipes connect the outlet of a safety valve, incorporated in the valve chest, directly with the condenser.



Elevation of Russian mercury-vapor turbine layout

Another interesting layout likewise prepared by the Soviet Central Boiler and Turbine Institute, provides for the double-flow turbine exhaust to a single condenser, the steaming coils of which form part of a La Mont steam boiler circuit.

Heat Transfer by Jet Impingement

Writing in VDI *Zeitschrift* of October 21, 1939, Prof. H. Voigt and Dipl.-Ing. F. Franklin relate the results of investigations on the relative effectiveness of heat transfer by impact of steam jets on metal surfaces as compared with that in the common form of closed heater in which steam flows over the tubes and water through them. It was found that the velocity of water through the tubes has less influence on heat transfer in the latter case than in the former.

A number of tests were made with vertical concentric copper tubes, the inner tubes, of 16 to 18 mm, carrying the water and the outer tube, of 24 to 30 mm, having perforations through which the steam impinged in jets on the inner tube. The orifices or nozzles were aligned in groups of four at each level, and spaced vertically on 30 mm and on 60 mm centers. Impingement of the steam in this manner apparently broke up the condensing sur-

face film on the outside of the inner tube and thereby reduced the resistance to heat flow, for heat transfer rates ranged from 3000 to 11,000 Kcal/m²/hr/°C, (equivalent to 615 to 2250 Btu per sq ft per hr per deg F) depending upon the steam pressure, water velocity and other factors, as compared with 2000 to 3400 Kcal/

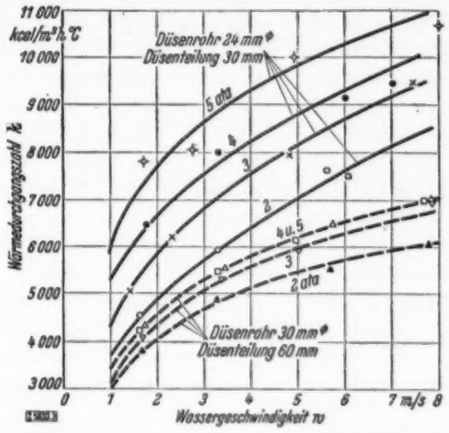


Fig. 1—Variation of heat transfer coefficient with water velocity, for different pressures and nozzle spacings

düsenrohr (jet tube); düsenteilung (jet spacing); wärmedurchgangszahl (heat transfer coefficient from steam to water); wassergeschwindigkeit (water velocity)

m²/hr/°C (410 to 697 Btu per sq ft per hr per deg F) without the use of jets.

In studying some of the other factors affecting the heat transfer rate it was found that the coefficient,

1. Increased with both steam pressure and water velocity when jets were employed;

2. Increased with decreased spacing of the jets, with decreased diameter of the outer tube (within certain limits), and with decreased distance between the outer and inner tubes, the optimum distance being 3 to 4 mm;

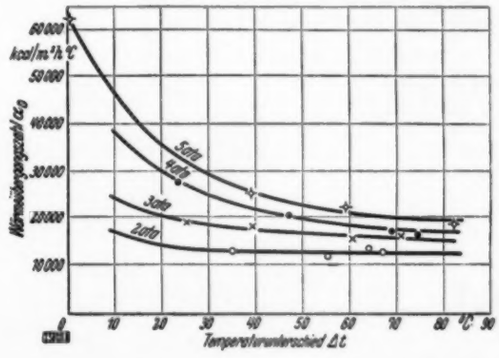


Fig. 2—Variation of heat transfer to tube wall with temperature difference for four pressures

wärmeübergangszahl (heat transfer coefficient from steam to wall); temperaturunterschied (temperature difference)

3. Varied to some extent with the diameter of the nozzle orifices, according to the pressure, but in general was greatest with orifices of 1.5 to 2.5 mm, other conditions being equal.

The diagram, Fig. 1, reproduced herewith shows the heat transfer coefficient plotted against water velocity, when using jets, for several pressures and two different arrangements of vertical nozzle spacing and nozzle tube diameters.

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RECONDITION YOUR OLD CONDENSER TUBES

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Write for this Bulletin

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The heat transfer coefficients from steam to tube wall were also calculated and plotted against temperature difference. This, as indicated in Fig. 2, shows a rather rapid drop in the lower temperature difference range with a flattening of the curves from 40 to 80 deg C. The coefficient is shown to increase with pressure, as would be expected.

Commenting on the results of the investigation, the authors observe that by employing jets a considerable reduction can be made in the amount of heating surface, and further, by using cheaper and thinner perforated outer tubes instead of copper or brass the cost of such a heater can be materially reduced. Further experiments are being made.

Steam Distribution System in South Wales

An unusual overhead steam distribution system supplying various industrial customers in South Wales, from the Upper Boat Generating Station of the South Wales Power Company, is described in the January 1940 issue of *The Power and Works Engineer* (London). This comprises a main pipe line over 6000 ft long, capable of carrying 100,000 lb of steam per hour at 350 lb pressure and 750 F, which supplies a series of distribution grids involving three pressures through automatically controlled reducing and desuperheating stations. Each such station serves an isolated low-pressure grid of about 1800 ft consisting of a steam ring-main with a condensate return pipe. The service lines are taken off these secondary grids. Not less than 80 per cent of all the steam delivered by the power company must be returned as condensate, to convey which there is a return main supported immediately below the supply line.

Throughout its entire length the supply line is carried on steel trestles fitted with roller supports. The total expansion, amounting to approximately 30 ft, is accommodated by expansion bends at 300 ft intervals.

Insulation comprises an initial coat of plastic asbestos compounds, a layer of 80 per cent magnesia, and a coating of hard-setting cement, the whole being sheathed with a casing of galvanized steel and painted. Drainage is effected by thirty 1-in. steam traps of the lifting type which discharge to the condensate return main. The condensate pipes are not insulated.

In individual cases where factories cannot meet the condition of returning the specified percentage of condensate uncontaminated by process use, evaporators are installed to supply their low-pressure steam.

Small consumers are charged a flat rate of 3*s* 6*d* (equivalent to about 80 cents) per 1000 lb of steam, whereas large consumers pay on a sliding scale involving maximum demand.

A Correction

Due to an error in the data submitted for the table on high-pressure central-station steam generating units, in the January issue, pulverized anthracite was given as the fuel for the Lake Road Station of the City of Cleveland. This should have been pulverized bituminous coal.

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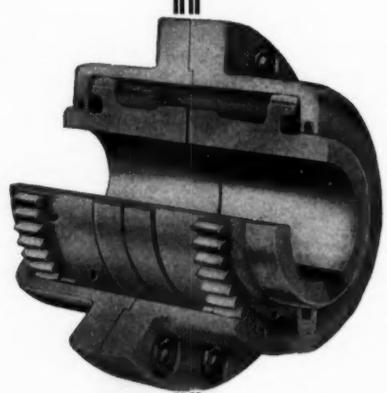

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EQUIPMENT SALES

Boiler, Stoker, Pulverized Fuel

as reported by equipment manufacturers to the Department of Commerce, Bureau of the Census

Boiler Sales

	1939 Water Tube No.	1939 Water Tube Sq Ft	1938 Water Tube No.	1938 Water Tube Sq Ft	1939 Fire Tube No.	1939 Fire Tube Sq Ft	1938 Fire Tube No.	1938 Fire Tube Sq Ft
Jan.....	75‡	380,903‡	52	201,151	50‡	64,511‡	35	42,752
Feb.....	70	309,235	48	185,257	45	58,028	45	55,173
Mar.....	98‡	386,132‡	58‡	238,830‡	58	54,752	50	49,039
Apr.....	62	276,507	48	195,910	36	42,177	37	52,421
May.....	104	406,828	60	330,653	74	96,037	61	68,288
June.....	99	486,631	58	190,242	52	58,793	63	86,975
July.....	88	320,901	67	271,561	59	66,595	69	98,074
Aug.....	88	397,093	56	190,762	52	55,714	53	69,494
Sept.....	136	715,911	45	169,241	74	110,082	58	62,794
Oct.....	118	596,509	51	191,932	58	71,060	46	48,231
Nov.....	80	518,654	55	198,589	66	75,318	38	39,198
Dec.....	54	222,861	67	335,297	39	36,906	42	59,821
Jan.–Dec., Inclusive..	1,072	5,018,165	665‡	2,699,425‡	663	789,973	597	732,260

‡ Revised.

Mechanical Stoker* Sales

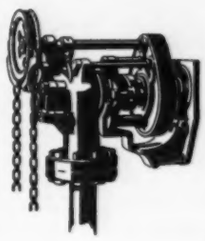
	1939 Water Tube No.	1939 Water Tube Hp	1938 Water Tube No.	1938 Water Tube Hp	1939 Fire Tube No.	1939 Fire Tube Hp	1938 Fire Tube No.	1938 Fire Tube Hp
Jan.....	44	17,067	28	9,484	145	17,842	76	10,991
Feb.....	46	20,715	36‡	12,450‡	140	18,217	76‡	12,216‡
Mar.....	46‡	19,068‡	54	18,820	122	15,743	52	9,434
Apr.....	50	18,888	35	12,698	114	13,652	71	11,058
May.....	62	29,245	32	10,830	153	20,010	106	15,342
June.....	83	34,618	28	9,284	184	21,801	166	21,378
July.....	66	20,893	45	17,449	213	30,829	191	24,816
Aug.....	65	22,589	35	10,991	311	41,310	269	33,199
Sept.....	94	40,983	47	16,250	345	45,731	279	28,780
Oct.....	63	23,480	42	15,809	313	39,784	300	44,111
Nov.....	58	24,307	27	8,151	208	27,428	201	26,382
Dec.....	44	18,263	47	17,617	163	20,775	172	22,500
Jan.–Dec., Inclusive...	721	290,116	456	159,833	2,411	313,073‡	1,959	260,207

* Capacity over 300 lb of coal per hour. ‡ Revised.

Pulverizer Sales

	1939 Water Tube No. N.†	1939 Water Tube No. E.†	1939 Water Tube Lb Coal/Hr	1938 Water Tube No. N.†	1938 Water Tube No. E.†	1938 Water Tube Lb Coal/Hr	1939 Fire Tube No. N.†	1939 Fire Tube No. E.†	1939 Fire Tube Lb Coal/Hr	1938 Fire Tube No. N.†	1938 Fire Tube No. E.†	1938 Fire Tube Lb Coal/Hr
Jan...	10	—	79,000	5	—	40,500	—	—	—	1	—	1,000
Feb...	7	—	89,600	7	1	38,020	—	—	—	—	1	800
Mar...	26	—	483,560	—	2	26,100	—	—	—	—	1	700
Apr...	8	—	155,050	2	2	26,200	—	—	—	—	—	—
May..	19	—	401,800	5	2	33,690	—	3	1,800	—	—	—
June..	19‡	1	297,230‡	7	2	49,440	—	—	—	—	—	—
July...	6	—	43,770	3	1	23,000	—	—	—	—	—	—
Aug...	3	4	53,700	8	5	155,390	—	—	—	1	—	1,000
Sept...	33	11	556,430	2	1	58,500	1	—	1,250	—	—	—
Oct...	26	2	419,410	2	—	7,650	1	—	2,800	—	—	—
Nov...	22	—	385,750	7	2	139,800	—	3	3,750	—	—	—
Dec...	3	—	19,200	15	1	208,700	—	—	—	—	—	—
Jan–Dec., inclusive.	179‡	18	2,930,500‡	63	19	807,390	2	6	9,600	2	2	3,500

† N—New boilers; E—Existing boilers. ‡ Revised.



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Beco-Turner cross baffles increase efficiency and capacity. They change the flow of gases from parallel flow to more effective cross flow. Adaptable to any vertical bent tube boiler.

For low-head boilers

Give cross-flow effect. Can be installed at any angle and in any design to give greatest efficiency.

Upon receipt of blueprints showing your existing boilers, our engineering department will gladly prepare drawings showing any possible improvements in baffle design.

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Ruggedly constructed, without parts subject to rapid wear, a Dean Cleaner stays on the job without constant repairing.

DURABILITY + THOROUGHNESS

In every type of boiler a Dean Cleaner will remove all incrustation, from both sides of the tubes, and stay in good working condition for years. This means that the cost per each cleaning can be figured in pennies.

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WRITE FOR BULLETIN #133

TUBE CLEANER

The WM. B. PIERCE CO., 51 N. Division St., Buffalo, N. Y.

Alex Dow Retires

Retirement of Alex Dow, for many years President of The Detroit Edison Company, was announced on February 5, although he will continue as a Director and as Chairman of the Executive Committee of that company. He will be succeeded by Alfred C. Marshall, who has been Vice President and General Manager of the company since 1923.

Born in Glasgow, Scotland, in 1862, Mr. Dow came to this country in 1882, first taking a position with the B & O Railroad and later with the Brush Electric Co. As City Electrician of Detroit in 1893, he built the municipal plant and became associated with the Edison Illuminating Company, predecessor of The Detroit Edison Company, in 1896. He has long been a recognized leader in the utility field and was the recipient of many engineering honors, including the Presidency of The American Society of Mechanical Engineers in 1928.

Mr. Marshall is a graduate of the University of Michigan and, after spending several years with the Detroit Public Lighting Commission and later in the electric railway field, entered the employ of The Detroit Edison Company in 1904 as construction engineer. He left in 1905 to become associated with the Port Huron Light and Power Company, later becoming assistant to the president of the Eastern Michigan Edison Company, but returned to The Detroit Edison Company in 1913. He was made Vice President in 1923.

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BALANCED VALVE-IN-HEAD

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